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● **H. Wood** *is the man who developed the Farnboro Indicator while in charge of engine testing work at the Royal Aircraft Establishment, Farnborough, England. After having left in 1925 to engage in engine, fuel, ballasting, heating and ventilating work on the airship R-101 at Cardington, he returned to the R.A.E. in 1928. He had started with the R.A.E. in 1914. He began to concentrate his efforts on liquid-cooled engines when he went with the Air Ministry in 1931. While there he worked on the Schneider Trophy engine. He joined Rolls-Royce as experimental engineer in the aircraft division in 1934. He made a tour of aircraft centers in the United States before returning to England.*

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Engine Types and Requirements for Preparation of Fuels

By F. C. Mock
Bendix Products Corp.

THERE are two main requirements as to proper preparation of the fuel charge for rapid combustion in our present engines:

(1) The fuel must be vaporized, or in a similarly small order of subdivision, before ignition.

(2) The fuel and air must be intimately mixed.

Light fuels, such as our present aviation gasoline, may be vaporized in the carburetor and supercharger to quite a satisfactory degree, providing that the intake air is heated when flying at low temperatures. With this system, as we know, the air and fuel mixing is quite thorough.

Heavier fuels, if released in the carburetor, may not vaporize in the intake-air flow, but instead may puddle and trickle on the side walls. Under such conditions, not only does the fuel fail to reach the cylinders in metered charges but also, if and when it should do so and if it vaporizes in the cylinder rather than getting on the cylinder wall, there is usually inadequate means for mixing the vapor with the air charge.

Fuel injection takes care of metering and delivering the fuel charge but, in practice with any jets the writer has seen, the fuel is delivered directly to what really is only a small part of the volume occupied by the air charge. Not only is the fuel not mixed, but it is so concentrated that the heat transfer from the air charge, which is an essential part of the vaporization process, is unduly slow. To deal with these difficulties it has been necessary to provide special charge turbulence or swirl for best results with nearly every form of injection.

Even though such turbulence be applied, the matter of time for its effective operation is important. Fuel sufficiently volatile to vaporize in the cylinder on the intake stroke, if injected then, apparently can be mixed completely with the air charge before the spark occurs. In air-cooled engine practice this condition seems to be possible with fuels of maximum boiling point of around 450 deg. fahr. With higher volatility limits, the combustion-chamber and wall temperatures do not seem high enough to insure vaporization on the intake stroke, and adequate temperatures are not reached until late in the compression stroke. With spark ignition the proportion of the air charge that can be reached by fuel *vapor* is critically dependent upon the fuel volatility, accurate injection timing, and the degree of turbulence obtainable.

MUCH effort has been, and is being, expended upon new engine types, fuels, and fuel feeds. It is submitted that a clear and fairly simple relation exists between the mutual requirements and limitations of these several developments, the understanding of which may be helpful in many ways.

Unfortunately for this presentation, both commercial and military restrictions make it impossible to describe in detail the experiments and tests upon which these conclusions are based, but they seem to have been supported consistently by the work of other investigators. I am rather certain that what this reasoning indicates may be relied upon as to what is going to be possible or impossible in future engine development along the lines described.

Long experience with many different types of fuel feeds indicates that there are two main requirements as to proper preparation of the charge for combustion in our automotive high-speed type of engine:

This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, W. Va., June 5, 1936.

(1) The fuel must be vaporized, or in a similarly small order of subdivision, before ignition.

(2) The fuel and air must be intimately mixed.

Vaporization Necessary for Rapid Combustion

Every instance that I have observed in nature supports this view; candle and kerosene wick flames, and the combustion of wood and coal are common examples. The idea, often advanced until recently, that the fuel-oil spray in compression-ignition engines burnt completely without preliminary vaporization has now been controverted by the very instructive photographic studies of the National Advisory Committee for Aeronautics. Even when the combustion rate is relatively slow, it seems that molecular separation, by heat or catalytic action, is an essential step.

Necessity for Homogeneous Mixing

The foregoing seems to be a primary requirement for obtaining maximum power, that is, burning of the whole air charge, and for obtaining minimum fuel consumption by burning efficiently the whole fuel charge. In addition, it has an important effect, with high-speed spark-ignition engines, upon the rate of flame propagation after the initial spark.

With compression-ignition engines, another important effect of this requirement is in the matter of heat exchange, as governing the temperature necessary for vaporization. At 90 per cent volumetric efficiency of the air charge and 15:1 compression, a volume of air 15 times the weight of a drop of fuel will be a sphere of air about 9 times the diameter of the fuel drop. If we assume the fuel drops enter the cylinder at a temperature of 200 deg. fahr. and must reach a temperature of 700 deg. fahr. to vaporize, the heat-exchange computation indicates a drop of 90 deg. fahr. in the temperature of this air sphere, that is, the air charge must be over 790 deg. fahr. to achieve ignition.

If time is not available to complete the heat exchange with this volume of air, say for instance that the exchange is completed with only half this volume of air, then twice the temperature differential will be required, that is, the air will have to be at 880 deg. fahr. in order to achieve vaporization before ignition.

A similar temperature requirement will exist in localities of local concentration of fuel sprays, for example when the air/fuel weight ratio is 7½:1. This is clearly the reason why the N.A.C.A. photographs show ignition beginning at the edge of the fuel sprays, and indicates that the air compression required for initial ignition should be lower as the drops are smaller. It also suggests that the effect of increased turbulence upon initial ignition might be uncertain as it would affect both the fuel concentration and rate of heat exchange. After the initial ignition, however, increased turbulence should increase both the rate and completeness of combustion.

In our earlier carburetor work we rather generally assumed that, if we could only meter any fuel correctly in the carburetor and distribute it equally in the intake manifold, that all would be happy and efficient ever after. The real awakening came when we first tried injecting fuel and found that it was not easy, even with a volatile fuel, to equal the consumption rates of the carburetor engine. We later found how to deal with this difficulty, but the evidence of a large number of different experiments has accumulated to show that liquid fuel is present and inert in the cylinder much more often than is generally supposed; that introducing properly metered fuel charges into the cylinder by no means insures efficient combustion; and that thorough mixing of the air and fuel charges is essential if we are to improve upon our present standards of operation.

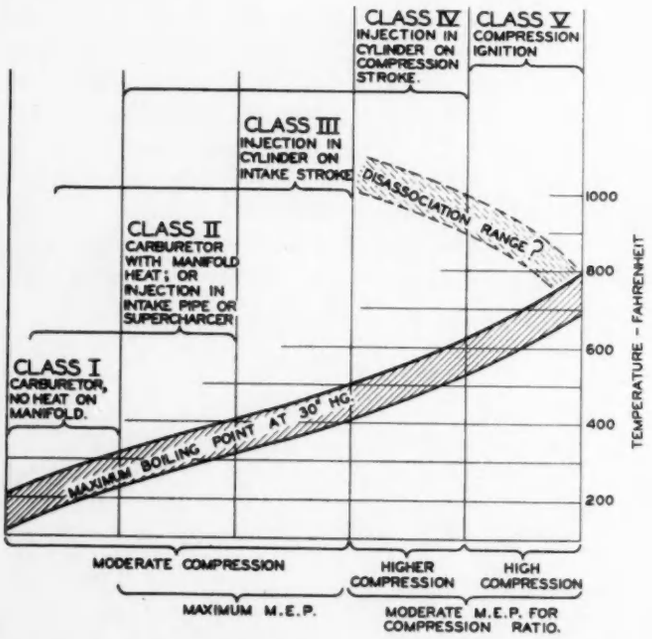


Fig. 1—Fuels Graded by Volatility, and Engine Types Suitable for Their Use

Engine Types Determined by Fuel Volatility

Putting together these two requirements of vaporization and mixing, we obtain the relations of fuel volatility and engine type shown in Fig. 1 which, it will be noted, embody pretty nearly the full substance of this paper. Though the boiling points of the heavier elements of the fuel, here used as an index, are only an indirect indication of its mean volatility, they do indicate a trend; also they largely govern its behavior on a "hot-spot" heating surface.

(Class I) Fuel Boiling below 280 deg. fahr. – This group of fuels has long been dealt with easily since they vaporize readily in the carburetor and manifold at most sea-level atmospheric temperatures; the air and fuel vapor usually mix quite well on the way to the cylinder.

(Class II) Fuel Blends Boiling between 140 and 360 deg. fahr. – These fuels may be used efficiently with a carburetor and heated intake manifold, which heating is responsible for some slight power loss and an increased tendency toward detonation. They also may be used without such loss: (a) by injection into the properly designed supercharger of a radial engine; (b) by timed injection into the intake pipes of nearly any form of engine; (c) and by injection into the cylinders. Special means of mixing the air and fuel usually are needed with the two latter forms.

(Class III) Fuel Blends Boiling between 200 and 450 deg. fahr. – This fuel class, adapted to injection into the cylinder on the intake stroke, receives its upper limit from the requirement that the combustion-chamber walls upon which the spray strikes must be higher in temperature than around the 90 per cent boiling point of the fuel. This condition is in order that the fuel may be vaporized or fogged immediately upon injection, without waiting for the higher temperatures at the top of the compression stroke; thus, the remaining time-portion of the intake stroke and the whole of the compression stroke are available for diffusion and mixing of the charge.

Special means for augmenting natural charge turbulence usually are required.

(Note: These same fuels may be used with a carburetor by a very considerable application of heat to the intake charge but, although perhaps suitable for other automotive service, this expedient, because of the ensuing power loss and detonation tendency, seems less desirable for aircraft use.)

Only with these three first classes have we found it possible to burn the full air charge of the cylinder and to realize the full mean effective pressure of which the engine is capable. As would be expected, when this result is achieved, the detonation tendency depends directly upon the octane rating of the fuel, which rating thus limits the compression ratio, cyclic efficiency, and power output.

(Class IV) Fuels Boiling between 300 and 560 deg. fahr. – Further rise of the fuel boiling point results in incomplete vaporization if the spray be injected against the piston or cylinder-head on the intake stroke. It is therefore necessary with such fuels to inject late in the compression stroke (see Fig. 5-A) and also to raise the compression ratio above 6:1, in order to inject into the air temperatures necessary to insure a dry mixture. As will be seen by reference to Figs. 5 and 6, while this condition involves higher compression ratios than ordinarily can be reached with the more volatile fuels without detonation, it nevertheless permits pressures materially below those necessary for compression ignition. The very short time available for diffusion of the fuel vapor in the air seems to prevent burning the full air charge, and I have never seen an engine of this type develop the mean effective pressure, in proportion to its compression ratio, that can be obtained with engines of the classes previously mentioned. In fact, it is usually difficult also to obtain as low a fuel consumption, in proportion to the compression ratio. All these factors, however, help to prevent detonation. A chief problem in this type of engine is obtaining a fuel-air mixture of proper igniting proportion around the spark-plug at the time of ignition.

(Class V) Fuels Boiling (Or Disassociating) above 560 deg. fahr. – This group of fuels seems only usable in the compression-ignition type of engine, which requires about 300 lb. per sq. in. higher compression pressure than Class IV fuels for regular dependable ignition under the varying loads of automotive service. Because, in my opinion, of the lack of time for mixing the fuel and air before combustion starts, this type of engine has so far been limited as to its mean effective pressure. Also it does not seem to realize the economy of fuel consumption of the first three engine classes described in proportion to its compression ratio, except at light loads. In its most successful forms, however, it seems to give the lowest fuel consumption of the group. It is sometimes forgotten that the fuels used in compression-ignition engines are usually deficient by about 10 per cent in B.t.u. per lb. as compared with the more volatile fuels now used with automotive engines.

Considerations for Different Fuel Volatilities

Aircraft engines thus far have been equipped to use mainly the Class I fuels, using a carburetor with no great application of heat to the charge; although, in the last few years, both hot-air and intake hot-spots have been fitted, chiefly to deal with ice formation. With the increasing requirements of high-altitude performance, greater supercharging, and the improved standard of performance set by recent developments in fuel injection, there has been initiated a radical change in carburetor form. The mixture proportion will be corrected automatically for both air density and temperature, and according to the cooling requirements of the engine under heavy or emergency load. The fuel-discharge jets will be brought nearer and nearer the supercharger, and the older methods of fuel metering, entirely adequate and reliable at fixed altitude and temperature, may be superseded by more or less elaborate automatic devices. Incidentally, formation of ice due to refrigeration from fuel evaporation will be minimized in this development. It is probable that this trend will blend these fuels into the next class following.

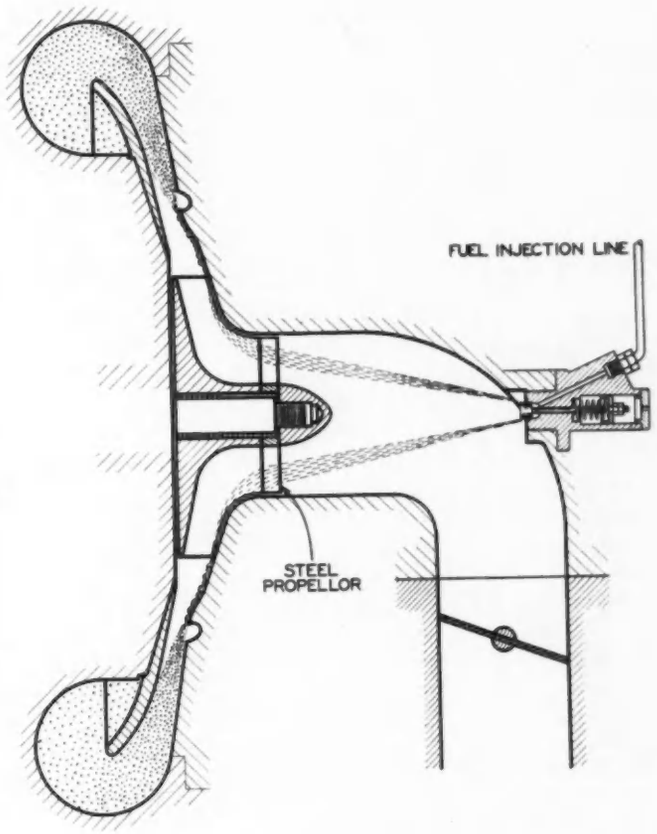


Fig. 2—Recommended Design for Injection into Supercharger

Fuel Spray Into the Supercharger.—For Class II fuels, this design in the writer's opinion, holds considerable promise toward giving maximum power, minimum fuel consumption, and general improved operation, while comparing favorably with the carburetor in simplicity and consistency in operation. Our experiments indicate, however, that certain minor changes in supercharger design sometimes may be desirable for best results.

In the first place, the fuel must be injected centrally into the supercharger; otherwise, it will not come out equally around the periphery. This condition is somewhat difficult to arrange when the supercharger drive shaft extends onward into a rear-accessory section of the engine, and probably requires multiple small orifices for fuel discharge, symmetrically spaced around the supercharger center.

Next, we have found that the stationary vane diffusers around the supercharger wheel should always terminate symmetrically with reference to the different intake outlets from the ring manifold or, in the lower range of operating

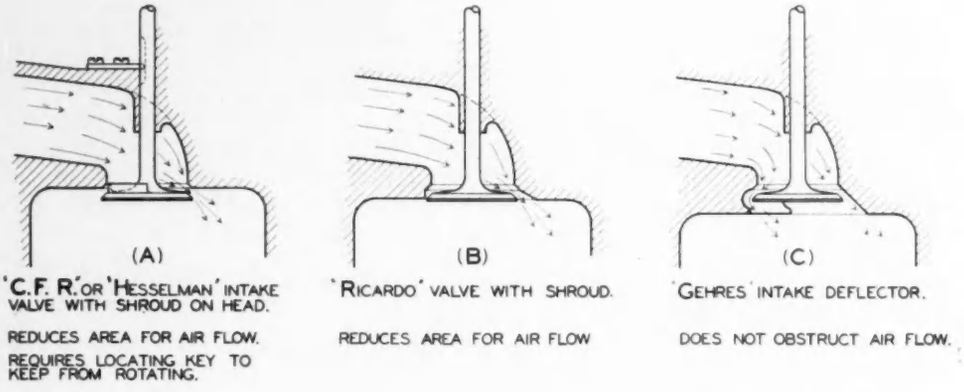

(A)
'C.F.R.' OR 'HESSELMAN' INTAKE VALVE WITH SHROUD ON HEAD.
REDUCES AREA FOR AIR FLOW.
REQUIRES LOCATING KEY TO KEEP FROM ROTATING.
(B)
'RICARDO' VALVE WITH SHROUD.
REDUCES AREA FOR AIR FLOW
(C)
'GEHRES' INTAKE DEFLECTOR.
DOES NOT OBSTRUCT AIR FLOW.

Fig. 3—Types of Intake Valve Which Deliver Air Unsymmetrically into the Cylinder Space, Thus Setting Up Swirl

temperatures encountered, the fuel, although atomized when leaving the wheel, may condense again when it strikes these diffusers and be washed along them leaving their outer tip in low-velocity trickles of coarse drops, which tend to pass directly into the nearest intake-pipe opening.

Fig. 2 shows a device for re-atomizing this trickle, also a type of diffuser which seems to give particularly good fuel distribution. It involves the disc- or flat-venturi type of expansion chamber, with a groove of re-entrant whorl and sharp edge terminating the inclined surface on which the fuel collects as thrown off the supercharger wheel. The fuel flies off this sharp edge into the passing air as a very fine spray, and this arrangement makes one of the most effective atomizers of large capacity that I have ever seen.

In the writer's estimation this type of supercharger diffuser should be more efficient than the vane type for the following reasons:

(a) With our present high-supercharge ratios the air comes off the wheel at almost the velocity of sound, a condition where it is particularly susceptible to surface friction. The mirror surface of minimum area, obtainable with the disc-type diffuser, has much less air friction than the cast and hand-scraped labyrinth passages of the vane type.

(b) Under the varying loads, speeds, and altitudes of aircraft service the air comes off the wheel at varying angles which variation, with a fixed-position vane diffuser, results in undue impact loss. With the disc-type diffuser the air can change its path without impact loss and, if properly designed, the only kinetic loss is that due to over-expansion, which is minimum.

(c) Consideration of the action of centrifugal superchargers shows that, due to the continuous circumferential acceleration as the air moves outward from the center, it must be compressed against the front of the blade and rarified at the back of each blade, so that the rear of each bucket space between blades delivers the main outward flow. Passage of these jets across the cavities of the vane-type diffuser produces in effect a siren, with continuous shock loss. A disc diffuser may be made with a converging contour just outside the blades which will equalize partially the pressure differences in the peripheral discharge.

One important consideration in the design of a disc-type diffuser is that the curves shall be gentle and comparable to what would be required for a streamline body intended to pass through the air at 800 ft. per sec. We have found that an attempt to widen the area too rapidly results in discontinuity and in the air-flow breaking away from the wall.

A point to be watched for in the design is that, when subjected to a sudden closing of the throttle at high engine speed, a collapsing force of some tons is exerted on the rear cover-plate; so that it is necessary to provide small bracing bosses or struts, which should be located as far out as possible from the rim of the impeller wheel.

Timed Injection Into Intake Pipes

This method, as generally applied, differs from the foregoing in that the fuel discharges are times definitely to occur through a certain duration on the intake stroke, instead of delivering continuously into a ring manifold as with injection into the supercharger. A major problem with this method is to insure that, throughout the intake stroke of a given cylinder, the fuel-and-air charge deliveries into the cylinder continue in equal ratio.

The air-flow in any intake manifold is in the form, on a time scale, of an irregular series of pulsations, whose period and profile are governed partly by the natural frequency of the intake manifold and partly forced by the suction stroke of the pistons. As plotted in degrees of crankshaft rotation, these pulsations of air-flow therefore vary according to the engine speed. Another irregularity is that, when the engine is partly throttled, the compression space in the cylinder is at approximately atmospheric pressure from the previous exhaust stroke, and higher in pressure than the throttled intake manifold; so that the first thing that happens when the intake valve opens is an outflow from the cylinder into the intake manifold, later to be followed by an inflow when the downward piston travel has brought the pressure in the cylinder below that in the manifold. Thus at sea level and at cruising power there may be no air-flow into the cylinder during the first 60 deg. of intake stroke.

On the other hand, with the same weight of air charge, at altitude, there will be slightly less intake pressure than at sea level, but a materially lower exhaust pressure and lower depression in the compression space above the piston travel, so that the reverse flow out through the intake valve will terminate sooner and the inflow to the cylinder begin much sooner on the intake stroke than at sea level.

Another difficulty encountered in trying to make the fuel charge keep pace with the air charge is that, while the fuel spray in suspension in the air stream may travel right along with it, a large amount of the fuel tends to deposit out on the bends and thereafter to travel toward the engine on the intake walls, lagging greatly behind the air. With an air-cooled engine this particular difficulty may be reduced if the fuel dis-

charge is directed to strike the intake pipe just about an inch outside the cylinder port, so that it quickly encounters the hot wall of the port. All this discussion points toward the use of some special means for insuring that mixing swirl or turbulence is present in the cylinder.

Fig. 3 shows three intake-valve constructions which give this effect:

(*A*) The "shrouded" intake valve, long used in Diesel practice and now popularized in the Hesselman and C.F.R. engines.

(*B*) A modification proposed by Ricardo, which avoids the necessity of holding the intake valve against rotation.

(*C*) A method proposed by Gehres, which has the advantage of allowing practically unobstructed air flow through the valve orifice, which the two former examples do not permit.

It may be pointed out that types (*A*) and (*B*) give the greatest tendency toward swirl at the beginning and end of the valve opening, while type (*C*) gives the highest swirl effect during the middle of the intake stroke, a matter that sometimes is important in connection with fuel-injection timing. A certain rotational effect also exists with the air-flow through the ordinary valve-and-head cylinder port at high engine speed, which effect may perhaps be augmented by flattening the port approach to the valve seat.

Rather peculiarly at first sight, these last-mentioned methods seem to exert very little additional resistance to air-flow into the cylinder, but this result seems logical when it is considered that the conversion of velocity flow into pressure energy may be more effective when the orifice delivery occurs tangentially into a whirl than when its energy is directed centrally and with shock into a mass having no particular motion. Somehow this comparison suggests the difference between packing a trunk by laying the contents in orderly layers, as compared with tossing them in indiscriminately.

In any case, of all of the efforts toward injection that we have made or that have come under our complete observation, only those have been consistently satisfactory that have employed special mixing turbulence. We have observed that, without special design, the incidental or accidental turbulence in a cylinder is sometimes nearly adequate at high engine speeds, but that incomplete mixing becomes in evidence as the speed is decreased. A fairly good comparison of different constructions as regards turbulence can be obtained by finding out at how low a speed each is effective.

Provided a homogeneous mixture is obtained in the cylinder, timed intake-pipe injection permits enlarging the intake-air passages with a consequent gain of power over the hitherto-used standard carburetor and intake manifold. This method also improves the fuel consumption to whatever extent has been lost by improper distribution with the carburetor and manifold system. It seems unlikely that trouble will be experienced with this system from formation of ice in the intake pipes. This form may prove preferable over injection into the supercharger for an engine having cylinder-blocks in line. All these conclusions are presented as applicable only to Class I and Class II fuels mentioned in the beginning of this paper—those which are sufficiently volatile to vaporize rather completely in the intake passage before entering the cylinder.

Cylinder Injection With Spark Ignition

Many of the basic requirements for this type of fuel feed are described in the published reports of the Massachusetts Institute of Technology and of the National Advisory Committee for Aeronautics. I think that it has been the hope of everyone engaged in this work that some ideal compromise might be found between the detonation limitations of the carburetor engine and the high pressure and low mean effective pressure of the compression-ignition engine, but it now seems fairly certain that complete combustion of the full air and fuel charges can be obtained only by having so homogeneous a mixture of fuel, vapor, and air that the detonation limitations of charge, pressure, and temperature are primarily those existing with the carburetor engine for similar fuels.

On the other hand, certain secondary conditions governing detonation are distinctly more favorable with cylinder injection than with the carburetor engine. No preliminary application of heat to the air charge is necessary, and the cooling effect of the fuel spray is found to be capable of definite localization; so that it is possible to cool the hot center of the piston or the exhaust valve to almost any extent desired, at the same time finding just the application of heat needed for vaporization of the fuel.

Cylinder injection is, therefore, particularly suitable to the fuels of Class III that are heavy enough to avoid fire risk in the airplane but whose boiling points fall within the limits of the hotter surfaces found in the combustion chamber or, as generally termed, "safety fuels." Cylinder injection is, of course, necessary for any fuel used with engines having a scavenging function that occurs during the intake stroke, whether such engines are of the four-cycle or the two-cycle type. Finally, cylinder injection seems to offer definite advantages in the minor practical details of service, such as easier starting, very prompt and powerful acceleration, and almost no tendency to back-fire. On the other hand, the need for mixing turbulence seems more definite with cylinder injection than with timed intake-pipe injection.

With these heavier fuels, special means are necessary for starting, of which the simplest is a small priming carburetor using gasoline to be employed for perhaps the first minute of running. After this priming the engine will fire (with these Class III fuels), provided that the mixture proportion is a bit rich. It seems necessary to use a rich mixture in the warming-up stage, perhaps to make about the same mixture readjustments with this fuel and injection as we have to do now with gasoline and the carburetor.

Some caution may be necessary in opening the throttle after a long glide because the fuel can load in a too-cool cylinder; I have had an engine go firing right along smoothly for ten or fifteen seconds after all fuel charge was cut off. When we stopped the engine by cutting off the fuel charge, we found a

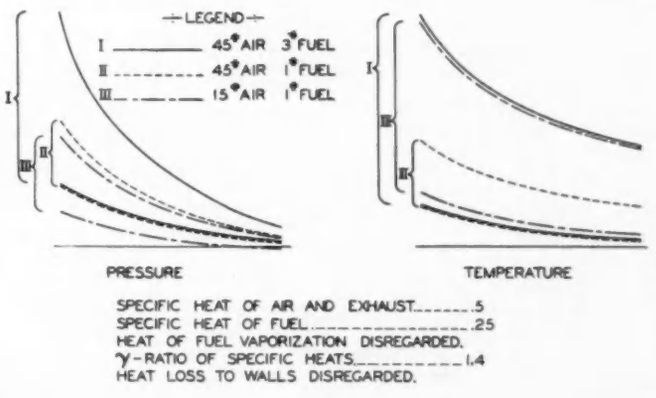


Fig. 4—Approximate Computation of Pressures and Temperatures with Throttled Vs. Unthrottled Air Charges, on Compression and Firing Strokes

layer of unvaporized fuel ⅛ in. deep on top of the piston, with no previous evidence of it when the engine was firing. Even with the engine warm, a combination of complete vaporization, adequate turbulence, and no over-rich mixture is necessary to obtain a completely clean exhaust; an enrichment of 2 to 4 per cent over the leanest setting for maximum power could make a very perceptible difference in the smell and opacity of the exhaust.

Possible Gain for Stratification.—It has been suggested often that a possible reduction in fuel consumption at cruising speeds might be obtained with cylinder injection if the fuel-vapor charge could be concentrated around the spark-plug to an extent that would permit a large excess-air content in the remote parts of the cylinder, for example, to permit cruising at one-half to three-fourths full-rated power, at full-throttle opening.

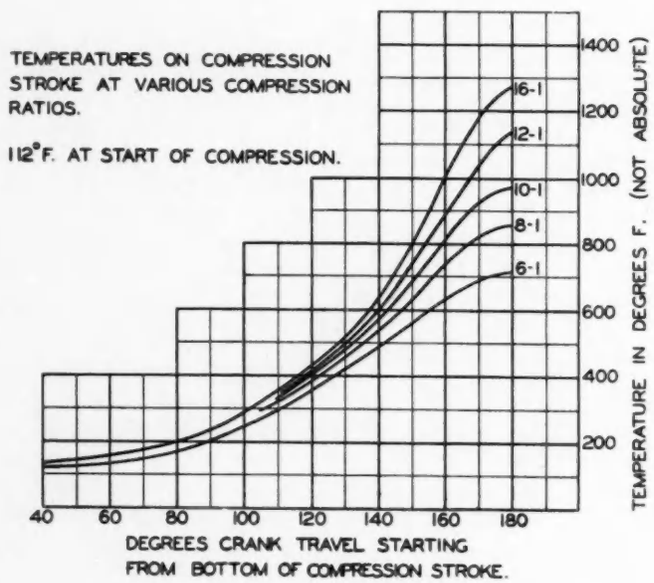


Fig. 5A—Temperatures on Compression Stroke at Various Compression Ratios

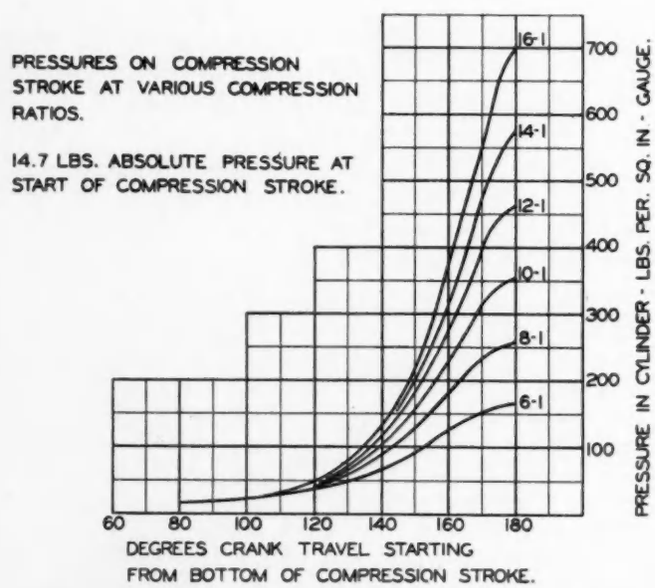


Fig. 5B—Pressures on Compression Stroke at Various Compression Ratios

While I have been able to approximate this condition, I have never improved the fuel consumption thereby, nor have I ever found any sound experimental evidence that the fuel efficiency of the modern high-speed mean effective pressure aircraft engine can be improved by adding a cool inert diluent to the charge. I have been able to operate engines smoothly on what appear to be homogeneous mixtures of the order of 21: 1 air/fuel weight ratio, also to idle on nearly full-open air throttle, but the fuel consumption per *horsepower-hour* was not quite as good as with around 16½: 1 air/fuel ratio.

Fuel Injection on the Compression Stroke, with Spark Ignition.—It will be noticed, in the classification of Fig. 1, that this group (Class IV) shares with the compression-ignition engine the imputation of being lower in mean effective pressure and power output than the classes earlier mentioned. I believe that this condition is due to their inability to burn all the air in the cylinder, even with the fuel feed increased to the point of a smoky exhaust. As shown in Figs. 5 and 6, at full load, temperatures adequate for vaporization are reached only in somewhere around the last 50 deg. of the compression stroke, and injection of a heavy non-volatile fuel before this time will result only in its partial collection in larger drops before the temperature rises enough for vaporization. Thus there is a relatively short time available for mixing the fuel vapor and air before the spark occurs.

It is the effort of the engineer, therefore, to obtain the earliest injection, the best diffusion of the fuel vapor and the air, and the best mixing turbulence that is compatible with maintaining an ignition mixture of not too high velocity around the spark-plug points. Also, some compensation for the presence of the excess air is found in the fact that the compression ratio may be raised with less detonation than would be experienced if all the air were burned; this condition tends toward lower fuel consumption, offsetting the possible losses due to the excess-air diluent and possible slow or incomplete combustion of certain parts of the fuel charge.

In order to get a basis of comparison and to establish a "par for the course," I would recommend that those experimenting with this type of engine try operating it with a carburetor and high-octane gasoline at, say 6:1, 8:1, and 10:1 compression ratio, with the throttle opened only to the detonation point and make "fish-hook" curves of the power vs. fuel consumption obtained through different fuel settings. Compare these curves with those obtained with injection, taking properly into account the B.t.u. per pound of the fuel. In my own experience the power per B.t.u. of the fuel always was indicated as higher with the carburetor than with the injection. Also, I believe that this tendency will continue if interpolated into the higher compression ratios and a comparison made with compression-ignition fuel and air consumptions.

From my own somewhat limited experience it seems that, if consistent ignition could be obtained, this type of engine would make at least as good a showing, on the basis of power output per engine and fuel weight, as the compression-ignition engine, but this showing will require that the fuel charge be segregated definitely in one part of the compression space near the spark-plug as in (*A*) Fig. 7, rather than having a condition such as (*B*) or (*C*). It will be noted that this arrangement reduces the detonation condition to the minimum, hypothetically, by having the flame sheet meet air alone at the last part of its travel.

To realize this condition in practice would require something like Fig. 8. The combustion should occur in a chamber separated from the piston chamber with walls hotter than the piston-swept walls, which latter must be kept cool to

maintain an oil film. Special means must be used for the mixing of air and fuel; the ignition point should be opposite the point of entry of air on compression but screened from direct air-current impact; and the profile of the combustion chamber should be such that spherical spread of flame spouted from the spark-plug points will not trap a body of combustible charge against a wall.

The problem of ignition is perhaps the most difficult of those encountered with this type of engine. Care must be taken not to get the spark-plug in the direct path of the spray as, at light loads, it may carbon up rather quickly. This difficulty is particularly marked in starting; either some minutes of running on gasoline, or the use of a burner something like the Packard "Fuelizer" of some years back, seem necessary for positive and consistent starts.

I might refer further to a few curiosities observed in trying to start with heavy fuels on the borderline of adequate temperature for vaporization. We found a condition where blue flame would come out of the exhaust ports while the engine was being cranked over, without enough power being developed to accelerate the engine's speed or make a noise when the exhaust valve opened. Repeating the injection and spark in the cylinder when laid on the bench would show stationary nuclei of blue flame remaining around the spark plug, but by no means complete ignition of the full spray charge.

Compression Ignition—Fuel Injection Engines

So much has been printed regarding this type of engine that it seems desirable only to repeat a few main conclusions, perhaps pointing out the similarity of the charge-preparation problem with this engine, to that with the types previously mentioned.

The first requirement, of course, is that of obtaining ignition, and the material of Figs. 4, 5, and 6 will show how this factor is dependent upon the compression ratio and the initial temperature of the air and fuel charge. These curves also emphasize the short time available for ignition advance before top-center if ignition lag is to be reduced by high-compression temperature. Our experiments painted a picture exactly consistent with the results recently shown by the N.A.C.A. photographs of combustion, and we concur thoroughly in their analysis and conclusions as to the sequence of actions taking place at this time.

Figs. 5 and 6 also emphasize the shortness of time for mixing, between the time the temperature is high enough for vaporization and the time ignition should take place. As one of our preliminary experiments, we built a rather elaborate "blow-torch," injecting a fuel-oil spray across a flat Bunsen burner flame through a small quiescent chamber with limited air charge, then into an expanding cone where further air was introduced tangentially under pressure. With this setup we got an impressive-looking flaming blast about three feet long, whose outer portion was purplish-blue and of which I was quite proud until one of the boys held a 2-in. square steel bar across it. Right out of the flame, drops of fuel oil condensed on the bar surface and ran down off the lower end in a clear transparent stream. Since that time I have been rather skeptical as to the completeness of combustion possible when the latter half of a Diesel fuel charge is injected into an already ignited mass.

Another interesting point demonstrated in our experiments was the relative unimportance of trick combustion-chamber shapes, as compared with the advantage of using high wall temperatures on the bushings, thimbles, or what not employed with these various precombustion-chamber types. From the fact that the best operation was obtained with metal temperatures far above the boiling point of the fuels used, we assumed that it was transmission of radiant heat, rather than conduction of fuel drops thrown on the walls, that was effective.

Without going into further detail, I would hazard the prediction that any compression-ignition engine successful in aircraft work will use a fuel oil carefully refined as to grit, acid content, heating value, and ignitibility; that it will be two-cycle, streamline uni-flow positively scavenged and supercharged, with short fuel lines; and that it will be either liquid-cooled or, if air-cooled, with careful temperature control and throttling at idle. Figs. 4, 5, and 6 may be worthy of some examination as to showing how the compression temperatures are raised and the pressure lowered by throttling. It is believed that more certain and consistent operation at light loads will be obtained by carefully coordinating

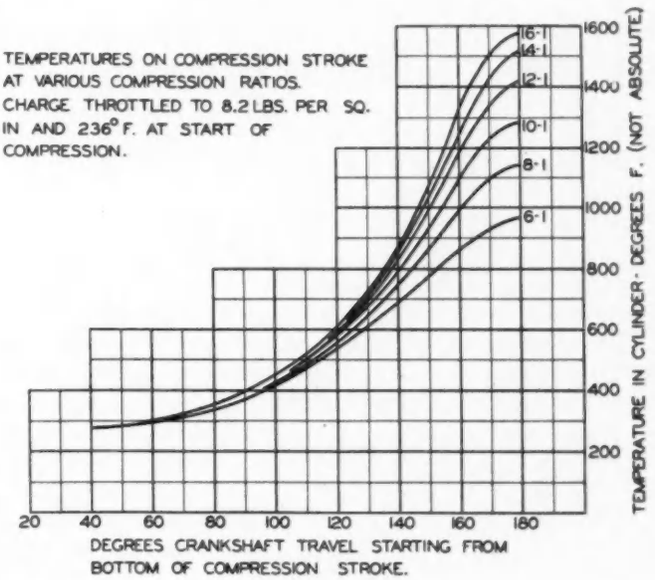


Fig. 6A—Temperatures on Compression Stroke at Various Compression Ratios

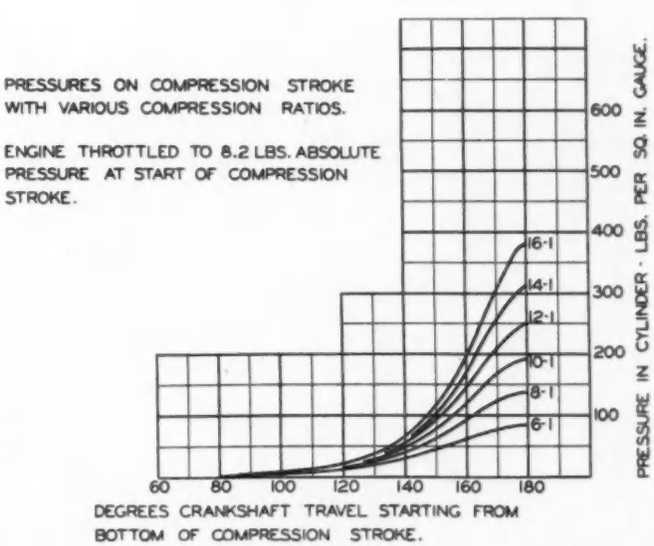


Fig. 6B—Pressures on Compression Stroke with Various Compression Ratios

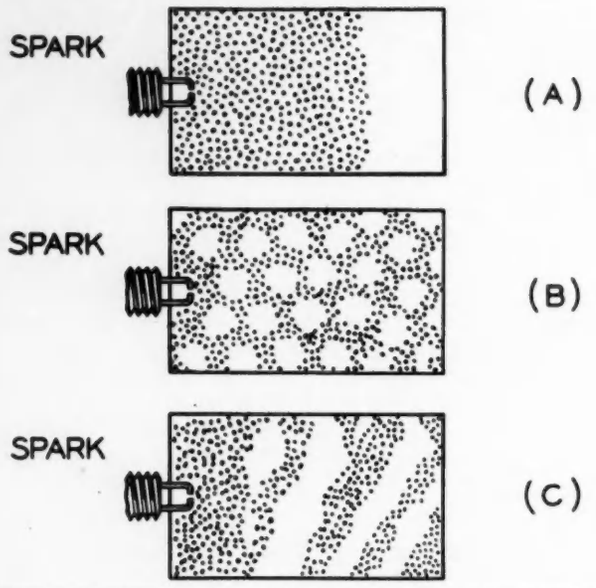


Fig. 7—Illustrates (*A*) Desirable, (*B*) and (*C*) Undesirable Distribution of Fuel Charge in High-Compression Spark-Ignition Cylinder

the intake-manifold pressure and fuel charge, just as in the spark-ignition engine. Since throttling the intake reduces the weight of new air charge without directly reducing the weight and temperature of the exhaust residue left in the cylinder clearance space at the start of the intake stroke, the heat exchange works out to give considerably higher temperatures throughout the compression stroke.

Conclusion

Referring back to Fig. 1, it seems that maximum power, for engine and short-cruise fuel weight, will always be obtained with Classes II and III, injection into the supercharger, intake pipe or cylinder, all on the intake stroke. From the viewpoint of low fire risk plus suitability to present aircraft service, Class III seems best. For long-range operation, the Class IV engine, with possibly Class II or Class III fuel, seems to the writer to offer the best compromise on power and weight, provided ignition problems can be solved; but it may not show much margin over the intake-stroke injection engine if 100-octane fuels become available, nor over the two-cycle supercharged compression-ignition engines if their various detailed problems prove capable of solution. See Fig. 9 for relative values.

For all engine forms, it seems to the writer that the rational and economically practical place to suit the fuel to the engine is in the oil refinery, not on the individual aircraft powerplant unit, and that we must always expect to use specially prepared fuels.

In any case, use of fuel injection in one of the different forms here described seems to the writer to be not alone an end in itself but an absolute necessity for further development of the aircraft engine. Immediate benefits available are increased power, reduction in fuel consumption, and diminution of intake ice. Cylinder injection will give improved local cooling of present hot-spots in the cylinder, perhaps to the extent of permitting materially larger displacements per cylinder than we now dare use. Close at hand is the ability to use a wider range of fuels and, beyond this point, lies the possibility of new engine forms and new cycles.

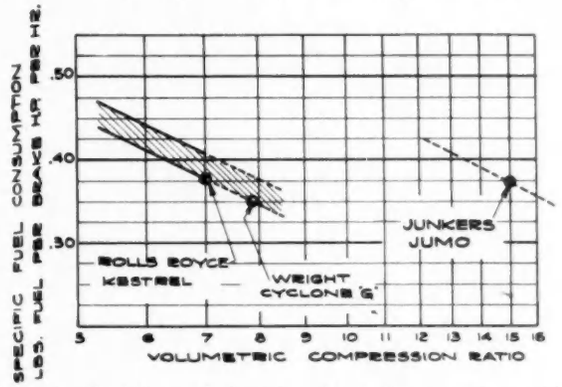

SPECIFIC FUEL CONSUMPTION IN LBS. FUEL PER BRAKE HP. PER HR.
.50
.40
.30
ROLLS ROYCE KESTREL
WEIGHT CYCLONE 'G'
JUNKERS JUMO
5 6 7 8 9 10 11 12 13 14 15 16
VOLUMETRIC COMPRESSION RATIO

Fig. 9—Variation of Cruising Fuel Consumption with Compression Ratio—Recent Spark-Ignition Vs. Compression-Ignition Engines

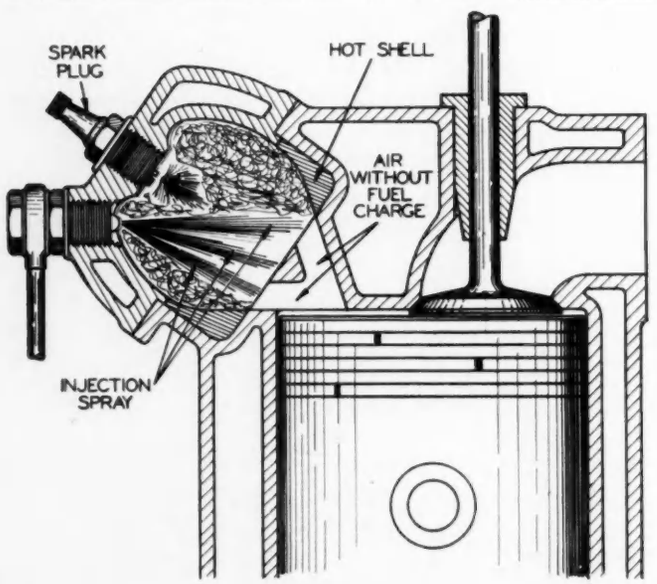


Fig. 8—Cylinder Type for Reducing Detonation; Injection Occurs Near End of Compression Stroke

Discussion

Effect of Injection on Volumetric Explained

—A. T. Gregory
Ranger Engineering Corp.

THE advantages to be gained by direct injection of the fuel into the cylinder, or by injection into the intake manifold immediately in front of the intake valve, must overcome the lower volumetric efficiency that these methods of fuel feeding give. The cooling effect of the vaporization of the fuel in the intake system for a carburetor engine, or for one with injection into the supercharger impeller, results in a higher charge density at the intake valve and, therefore, in higher volumetric efficiency, the improvement being approximately 8 per cent over the first two types of injection. The need for intercoolers is thus greatly reduced by mixing the charge before entry into the cylinder.

Two tests that illustrate this point were run on a Ranger engine equipped with a carburetor. The engine assembly for the two tests was identical as regards compression ratio, valve timing, and so on, with

the exception of the impeller gear ratio which was different for each test. The atmospheric conditions for the tests were very nearly the same. A manifold pressure vs. horsepower curve was plotted at 3000 r.p.m. for each test.

It was found that, whereas the engine delivered 460 hp. with the low gear ratio, it delivered only 390 hp. with the high gear ratio at the same manifold pressure. Computations of the horsepower required by the supercharger showed a difference of about 25 hp. between the two gear ratios.

The temperatures of the intake charges were measured by thermocouples laid on the outside wall of the manifold between the supercharger and the valve ports. These thermocouples were insulated carefully to reduce radiation effects.

The temperatures measured for the two tests described previously were 142 and 190 deg. fahr. respectively. Inasmuch as the manifold pressure was the same in the two cases, the charge density varied inversely as the absolute temperature. The loss in volumetric efficiency with the higher gear ratio was, therefore, 7½ per cent, the corresponding drop in brake horsepower being 35 hp.

Higher Pressures Predicted with Fuel Injection

—Gustave P. Toews
U. S. Naval Aircraft Factory

FUEL injection, as you have heard Mr. Mock point out, is an absolute requirement in furthering development of the aircraft engine.

This requirement, I would say, is because of the necessity for improved scavenging and cooling of the combustion-chamber interior to which the injection type readily lends itself, especially if the engine is supercharged. Such improvement should allow for increases in brake mean effective pressure of at least 50 per cent at specific-fuel-consumption values below those now required to obtain 200 lb. per sq. in. b.m.e.p. With what we know now I see no reason why aircraft-engine builders should content themselves with any value of mean effective pressure below 400 and possibly 500 lb. per sq. in.

The solution that fuel injection offers to the fire-hazard and carburetor-icing problems might be considered a real blessing to those engaged or interested in the aviation industry. It is for these reasons, in addition to the possibilities of higher power outputs, that the development of fuel injection seems well worth while.

Adaptability of "Safety Fuels" to Fuel Injection Discussed

—Major Edwin E. Aldrin
Standard Oil Development Co.

"SAFETY fuel," as presented several years ago by our company to the industry through cooperation stimulated by the U. S. Navy, had a boiling range of 300 to 400 deg. fahr., roughly. By dehydrogenation the octane number was 87 without lead. It had a flash point of about 110 deg. fahr., the same fire safety as kerosene. This fuel had to be injected into the cylinders to obtain the full value of its merits in economy and brake mean effective pressure. It was practical with certain existing engines if the carburetor manifold was heated. However, the full utilization of this fuel has had to wait upon fuel injection.

It was most interesting to note Mr. Mock's claims that fuel injection into the cylinders rather than into the manifolds was necessary for this class of fuel. Certainly on this basis, if experimental and test work is to progress to its full utility, effort should be concentrated on cylinder injection.

I have a suspicion that progress in injection will lead us to other new factors to consider in fuel as affecting efficiency besides octane number. Resurrection of the aromatic content or the like may be involved. I mention this point as illustrating the variations that will have to be borne in mind.

I have long wondered why the military services have not stressed more the importance of raising the flash point of fuel. This increase should reduce the fire risk in the air and on the ground. Some question as to complete safety in case of crashes exists, but this problem needs to be tackled from an all-inclusive standpoint.

The carburetor is becoming appreciated as a real automatic device, relatively simple. The problems of mechanical injection, although complex, do open a new field in which progress nevertheless may yield greater dividends in opening up new possibilities. If fuel injection provides a means for reducing piston temperature not otherwise obtained, the lubricating as well as fuel problem will be simpler, resulting in more possibilities in brake mean effective pressure.

For sake of completeness, it might be mentioned that jet propulsion as pioneered by Dr. R. H. Goddard might be added to the chart on Fig. 1 of Mr. Mock's paper. However visionary this subject seems now yet, from a fuel standpoint, it may permit the use of low-price fuels not otherwise wanted. The annual mention of the internal-combustion turbine and the use of steam in aviation might come in here too, permitting still another class of oil.

This paper reminds me of one I presented six years or so ago rating fuels for various aeronautical engines, before any knock-rating scale was in use or before cooperative engine testing of fuels was undertaken. Mr. Mock apparently is willing to risk himself as a target on questions that extend the borderlines of present activity.

Questions as to lubrication and corrosion of fuel pumps for injection purposes may be out of order to address to Mr. Mock, but possibly he can be more specific as to which line of progress is more likely to succeed—all things considered. From the fuel standpoint, what can we expect to be the next direction? Is a solvent oil in the fuel likely to be of value? Can different, ordinarily non-miscible fuels be metered and mixed prior to injection? If so, additional possibilities will be opened.

Means for Obtaining Mixture Homogeneity Reviewed

—Daniel P. Barnard
Standard Oil Co. of Ind.

THE author is certainly to be congratulated for this contribution to the understanding of the importance of mixture homogeneity in engine behavior—particularly where high specific outputs are required. His work shows most clearly how many imperfections, which could be made to appear unimportant in low-duty engines, cannot be tolerated when high performance really is to be obtained. All of which study suggests that it might be well to review briefly the accepted means for obtaining fuel disintegration and mixture homogeneity, particularly with carburetor engines.

Prof. George B. Upton, in the *Sibley Journal of Engineering*, January, 1923 issue, showed that the tearing effect on liquid particles due to differences in velocity between liquid and gas (for definitely turbulent conditions) was proportional to the square of this velocity differential. Because this value, in turn, is proportional to the pressure drop producing the velocity effect, it follows that particle size will tend to decrease (due to turbulence) in proportion to the pressure drop employed

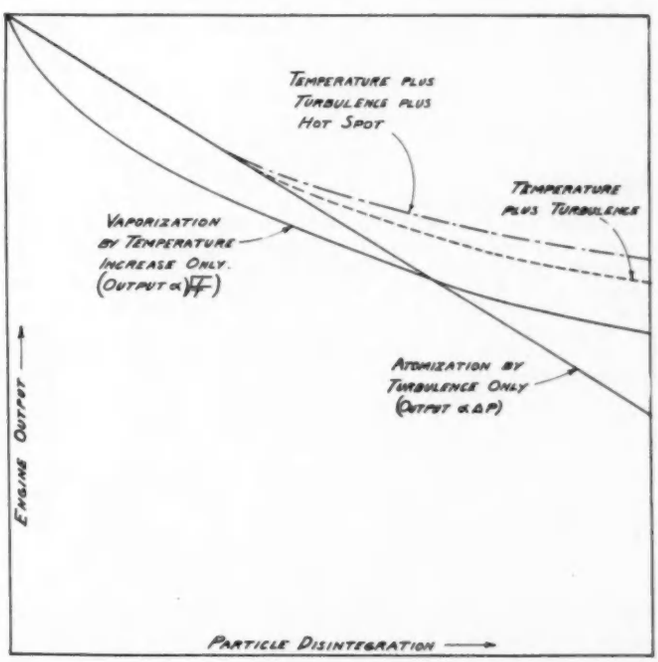

TEMPERATURE PLUS TURBULENCE PLUS HOT SPOT
TEMPERATURE PLUS TURBULENCE
VAPORIZATION BY TEMPERATURE INCREASE ONLY. (OUTPUT ∝ √ΔT)
ATOMIZATION BY TURBULENCE ONLY (OUTPUT ∝ ΔP)
ENGINE OUTPUT
PARTICLE DISINTEGRATION

Fig. *A*—(Barnard Discussion) Particle Disintegration Vs. Engine Output

to produce that turbulence. As manifold-pressure drop affects volumetric efficiency directly, the net result of obtaining atomization by mixture turbulence will be represented as shown by the straight solid line in the accompanying Fig. A. If, on the other hand, recourse is taken to mixture heating *only*, a curve such as the other solid line will result which shows that output will drop in proportion to the square root of the absolute mixture temperature required to produce particle distintegration.

These considerations lead to the conclusion that, if very little needs to be done to the mixture, it is probably best to depend entirely upon turbulence. If, however, a considerable amount of additional particle distintegration is required, it becomes more desirable to employ heat.

Actually the best combination may be represented by a line such as the dotted curve while, by the proper application of a hot-spot, a trend such as the upper broken curve may be attained. Finally, it is indicated that if the fuel is so recalcitrant as to require disintegrating energy which, if supplied by working through the entire mixture, would incur very large power-output losses, it will then become advantageous to resort to direct atomization of the fuel—that is, injection. Obviously where this condition occurs will vary with the standards of engine performance which must be met.

Engine Types Compared in Author's Closure

—F. C. Mock
Bendix Products Corp.

RELATIVE to comparison of these different engine types, it is obvious that such a comparison should be made under parallel conditions. Fig. 9, for instance, shows fuel-consumption values vs. compression ratios obtained with recent spark-ignition engines, and also a reported value for the Junkers Jumo. From this figure it will be seen that the spark-ignition engine will equal the specific consumption of the Junkers with the compression ratio of 7½ or 8:1; whereas the compression ratio of the Junkers must be around 15:1.

Ignoring various amounts of boost, rate of pressure rise and the like, it is apparent from comparison of Fig. A that, for the same *effective pressures*, the maximum pressures and engine stresses must be greater as the compression ratio is higher. It would seem to be an unavoidable indication, therefore, from Fig. 9 of the paper and Fig. A of this closure that:

(a) For a given power the compression-ignition engine, of an output and speed necessary for aircraft use, will be more highly stressed and heavier than a similar engine with spark ignition.

(b) That, for the same power output and *same* specific fuel consumption, the compression-ignition engine will be the more highly stressed and heavier.

The deficiency experienced with the compression-ignition engine as expressed in Fig. 9 must be due partly to the fact that its fuel is lower in B.t.u. per pound: to its high friction horsepower; and, in the opinion of the author, because of its inability to obtain complete vaporization and mixing before combustion. This criticism, it will be noted, is specific to heavier-than-air aircraft service.

Regarding the matter of air volumetric efficiency of the engine with fuel injection on the intake stroke, it will be noted that the charge density at the beginning of compression is a matter of the heat exchange between the exhaust residue, the new air charge, and the fuel charge. It seems in practice to make little difference whether this mixture and heat exchange are obtained partly in the intake pipe, or entirely in the cylinder on the intake stroke. Tests on one of the engines referred to reported a volumetric efficiency loss of a few per cent as compared with the standard carburetor engine, but also a slightly higher brake mean effective pressure for the injection engine at the same manifold pressure and temperature was recorded.

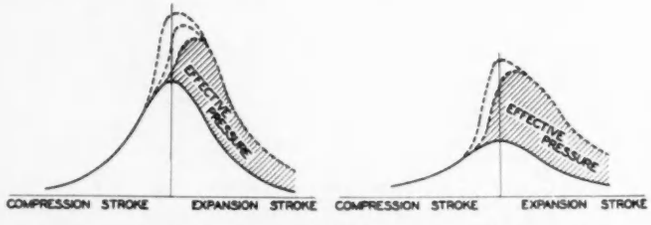


Fig. *A*—(Author's Closure) High and Low Compression Pressure—Showing Rudimentary Fact That, for Same Mean Indicated Pressure, Pressures Must Be Greater with Higher Compressions

Standards Committee Reports

At a regular meeting of the Standards Committee held at White Sulphur Springs, June 4, C. W. Spicer, General Standards Committee Chairman, presided. Reports as submitted by five Divisions were approved.

The Aircraft Division recommended several dimensional changes in the standard for Aircraft Bolts and Nuts, p. 558 and 559, 1936 S.A.E. HANDBOOK, bringing the S.A.E. Specification into line with the current Army-Navy Conference Standards used in both military and commercial air services.

The Ball and Roller Bearings Division recommended addition of a few dimensions to the present S.A.E. Standard for Ball Bearing Lock Nuts and Washers, p. 163 to 168, 1936 HANDBOOK, which will make this standard acceptable for use with either ball bearing or roller bearing installations. No modifications were made in the present standard for ball bearing lock nuts.

The S.A.E. Standard Method of Rating Fuels for Detonation, p. 446, 1936 HANDBOOK, was acted on in accordance with the recommendation of the Steering Committee of the S.A.E. Cooperative Fuel Research Committee, the note in the present specification referring to future octane rating being deleted.

The Production Division submitted several reports on machine tool standards which the Sectional Committee on Small Tools and Machine Tool Elements, for which the Society is a sponsor under the American Standards Association procedure, had submitted to the Society for approval as a sponsor. These were the proposed American Standards for Chucks and Chuck Jaws, for Lathe Spindle Noses and for Circular and Dovetail Forming Tool Blanks. In the case of the latter report approval was recommended, but it was also recommended that certain modifications bearing on the strength of these tools be given full consideration by the Sectional Committee before the report is submitted to the American Standards Association for final approval. The last report submitted by this Division relates to the standardization of twist drills in diameters ½ in. and smaller that is nearly completed in the Sectional Committee but has not yet been referred to the sponsors for their approval. The list of these drill sizes may be found on p. 682 and 683, 1936 S.A.E. HANDBOOK.

The latest addition to the family of Divisions of the S.A.E. Standards Committee is the Tractor and Equipment Division, appointed early in 1936. At its first meeting during the S.A.E. Tractor Meeting in Milwaukee in April, the Division recommended that several old tractor and industrial power standards adopted by the Society many years ago be discontinued as obsolete. These include Farm Engine Governors (p. 631), Stationary Engine Rating (p. 631), Water Pot or Hopper Capacities (p. 632), Driving Pulleys (p. 633) and Clutch Diameters (p. 633), all in the 1936 HANDBOOK. Other standards in the HANDBOOK such as those relating to Tractor Belt Speeds, Tractor Drawbar Adjustments, etc., are being brought up to date and will probably be reported together with new subjects before the Division, by the time of the Annual Meeting next January.

All of the Division recommendations referred to above were reported to the Council together with the action on them by the Standards Committee, and approved. As this was the final procedure in their adoption by the Society, the revisions and new data will be included in the 1937 edition of the S.A.E. HANDBOOK, to be published following the Annual Meeting next January.

Liquid-Cooled Aero Engines

By H. Wood
Rolls-Royce, Ltd.

THIS paper gives a brief resumé of the development of the Rolls-Royce Kestrel engine and then analyzes the requirements of the high-performance engine of the future, developing at least 1500 b.hp. and operating on fuels of high knock ratings.

The problems investigated include those of engine form, fuels, detonation, waste-heat disposal, cooling drag, cooling medium, and the mechanical and operational features.

Conclusions deduced from the arguments are:

(a) Compression ratios, charge density, and rotational speeds will need to increase and, therefore, cylinder bores and strokes will decrease; it may be necessary to adopt the sleeve-valve type.

(b) The arrangement of the engine will tend to multithrow crankshafts with more than two pistons per crankpin.

(c) It will be necessary to abandon methods of cylinder construction that rely on the art of the pattern-maker and founder, in order that cooling passages may be better controlled in cross-sectional area and more uniform metal sections are secured to reduce heat stresses and weight.

(d) The necessity for having complete knowledge of the fuel's performance in the actual aero engine requires a more exact method of assessing knock ratings.

(e) It will be necessary to liquid-cool the engine in order to deal with the increased waste heat, and to operate the cooling medium at high temperature in order to keep the cooling drag within reasonable limits.

WHEN the author was asked to prepare and deliver a paper to this meeting, he also was given freedom of choice of subject. As a member of a company that has built up a tradition for the manufacture of successful liquid-cooled engines and having in mind the great preponderance in U.S.A. of engines of the air-cooled type, it seemed appropriate to accept the invitation as a friendly challenge and to prepare a paper setting down our reasons for continuing with liquid cooling and our beliefs in the ability of this type to persist in the future when improved fuels will be available and when the problems of detonation and heat dissipation will be more intense than ever. We are conscious of the enormous strides made in the technique of air cooling, particularly in this country, and that the once-familiar jibe of cooling by petrol now no longer applies.

Similar progress has been made in England on the corresponding liquid-cooled problems, but the rate of progress has not been quite so rapid for various reasons. In America the air-cooled engine has always predominated and, when civil aviation started its rapid growth, it was logical to use and develop the well-established type. In England, on the other hand, civil aviation is comparatively small; most engines and aircraft are made for the British Air Ministry for military purposes; and balance between the two types of engine has been maintained to the technical advantage of both. Until very recently, however, the performance of the aircraft in which the engines were fitted was not such as to demand very great refinement in the reduction of cooling drag. Now that performances are improving, we have found it necessary to revise our views on the whole engine and aircraft problem, and we have embodied in this paper some of our arguments for your consideration.

SECTION 1

Kestrel Engines

Before developing the main theme of the paper it is appropriate to examine at this stage the progress that already has been made with a liquid-cooled engine that has been in production for eight years.

The construction of the Rolls-Royce Kestrel engine is familiar to most of you and will be described only briefly. The basic engine is a twelve-cylinder 60-deg. V-type of 5.0-in. bore, 5.50-in. stroke, having very small frontal area (see Fig. 1). In order to meet the varying demands of naval and military requirements the present series is produced in three main varieties, each of which is obtainable with one of three reduction-gear ratios; these gear ratios are 0.632, 0.553, and 0.477.

The S type is fully supercharged and is designed for operation in fighter and interceptor aircraft.

The MS type has a lower supercharger gear ratio and is used in flying boats and other aircraft requiring large take-off power, and whose operational height is of the order of 5000 ft. and where long range and, consequently, low consumption are essential.

The B or unsupercharged types are designed for bombers,

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 4, 1936.]

Table 1—Main Characteristics of Sub Types—Rolls-Royce Kestrel Engines

Type	Series	Normal Rating R.P.M.	Hp.	Altitude, ft.	Maximum Rating R.P.M.	Hp.	Altitude, ft.	Dry Weight, lb.	Compression Ratio	Rated Boost, lb. per sq. in.	Take-off Boost, lb. per sq. in.	Take-off R.P.M.
S.	IV-VI	2,500	600	11,000	2,900	640	14,000	955	6:1	1.5	6.0	2,240
M.S.	VII-IX	2,500	675	3,000	2,900	730	5,250	955	6:1	2.625	4.5	2,375
B.	X-XII	2,500	585	Sea Level	2,900	630	Sea Level	900	7:1	....	....	2,375
S.	XIV-XVI	2,600	690	11,000	3,000	745	14,500	955	6:1	3.25	6.0	2,225

Series	Piston Speed, ft. per min. Normal	Maximum	B.M.E.P., lb. per sq. in. At Normal R.P.M.	At Maximum R.P.M.	Hp. per Liter Normal	Maximum	Fuel Consumption (87-octane) R.P.M.	B.Hp.	Pt. per Hp-Hr.	Lb. per Hp-Hr.
IV-VI	2,290	2,660	146.5	135.0	28.25	30.2	2,500	485	0.515	0.476
VII-IX	2,290	2,660	165.0	154.0	31.80	34.4	2,500	548	0.465	0.430
X-XII	2,290	2,660	143.0	134.0	27.60	29.9	2,500	517	0.445	0.412
XIV-XVI	2,385	2,750	162.5	151.5	32.50	35.1	2,600	540	0.520	0.481

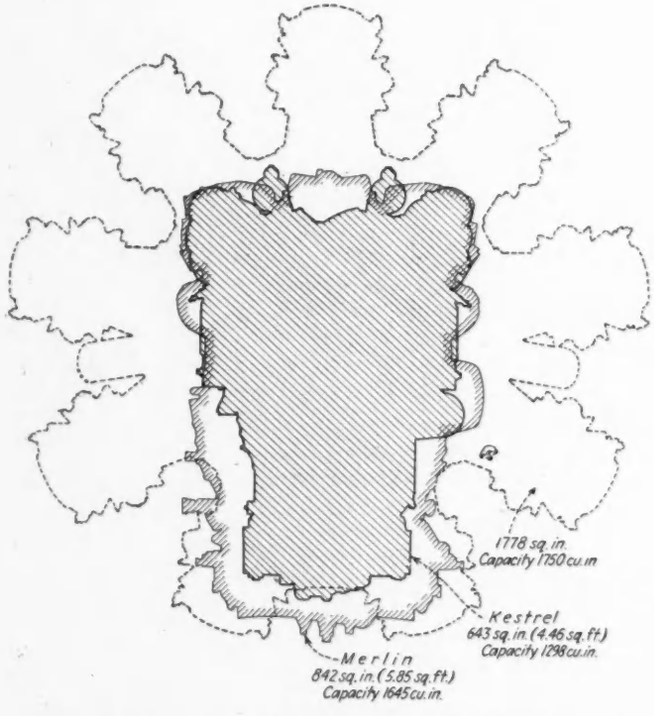


Fig. 1—Comparison of Frontal Areas, Liquid-Cooled and Air-Cooled Aircraft Engines

reconnaissance and cooperation aircraft where long range and, consequently, low consumption are essential.

In these engines we have produced a basic engine with a power output ranging from 585 hp. at sea level to 745 hp. at 14,500 feet or 730 hp. at 5000 feet. The main components of all Kestrel series are identical and interchangeable as between types, and the range of output has been achieved for a difference in engine weight of only 60 lb.

The main characteristics of the sub types of the series are shown in Table I.

It should be noticed that the preceding data have all been approved by Official British Air Ministry type tests. All the engines of the series also have been type tested for composite cooling, which allows the use of a radiator of 80 per cent of the frontal area demanded by normal water cooling. Under flight conditions where the heat to be dissipated is a maximum, the system is allowed to boil and the steam so generated is condensed in a small dragless condenser forming a part of the center section of the top plane. There is, therefore, an increase in the cooling per unit of drag and a reduction in the amount of interference drag.

All types of the Kestrel engine are fitted with a hand starter and have provision for driving two Constantinesco gun gears, two types of air compressor, electrical generator, petrol pump, revolution indicator, and for fitting a Hucks starter claw or a compressed-gas starter.

The supercharged types are fitted with an automatic boost control maintaining constant boost pressure up to the rated height. For take-off conditions a gate in the travel of the pilot's throttle lever puts the automatic control out of operation and substitutes a setting giving a positive boost pressure

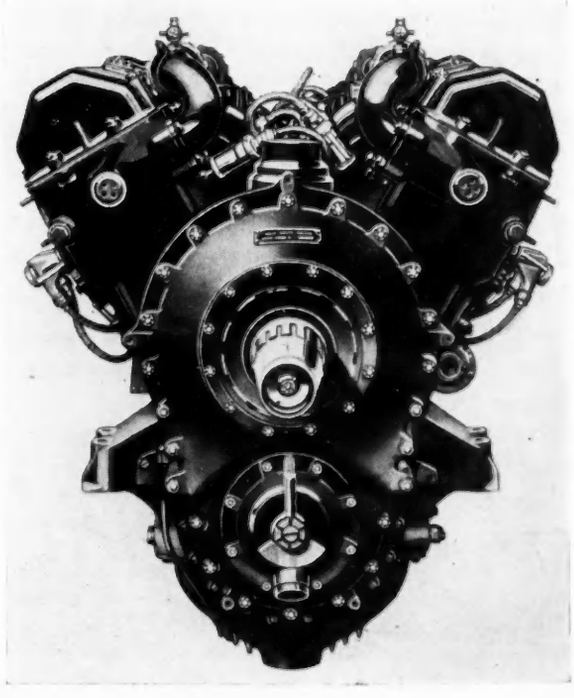


Fig. 2—Kestrel Engine (front)

of 6 lb. per sq. in. In emergency the automatic control may be cut out when the pilot has full control of the throttle.

Constructional Features

Cylinder-blocks are aluminum castings, having renewable valve seats screwed into the heads, bolted to carbon-steel cylinder liners through a soft-aluminum ring joint at the upper end and having special rubber rings at the lower end. Flanges on the liners bear on the crankcase, and the assembly

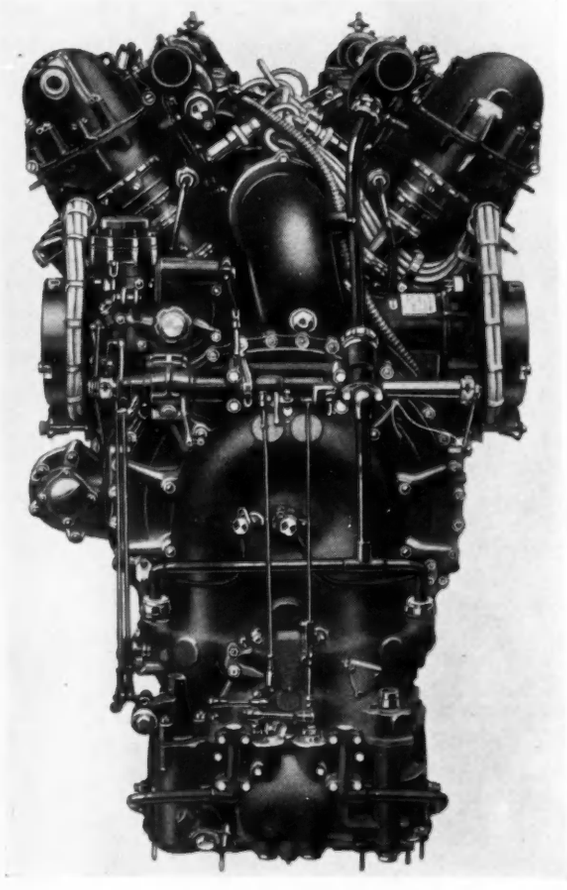


Fig. 3—Kestrel Engine (rear)

is held in place by long bolts enclosed in aluminum tubes. No cylinder walls are formed between the cylinders, and the lower end of the block is free to move over the liner.

Two inlet valves and two exhaust valves per cylinder are fitted, and the combustion space is a comparatively shallow disc. Exhaust valves are made of KE.965 steel stellited on the seat, and with sodium cooling in the stem.

Lead bronze is used for the seven main bearings and for the connecting-rod big ends. Each throw of the crankshaft has a pair of rods, one of which is forked and carries a steel-block big end metalled on its inside and outside cylindrical surfaces with lead bronze. The outer surface forms the bearing surface of the other rod which is straight.

The accessories are driven from the rear of the crankshaft by a torsionally flexible shaft, the rotation of which is limited by a hollow sleeve connected to its outer end by means of a damping clutch.

The reduction gear is a single-spur reduction bringing the airscrew shaft almost to the center of the front of the engine as this position facilitates the fitting of low-drag

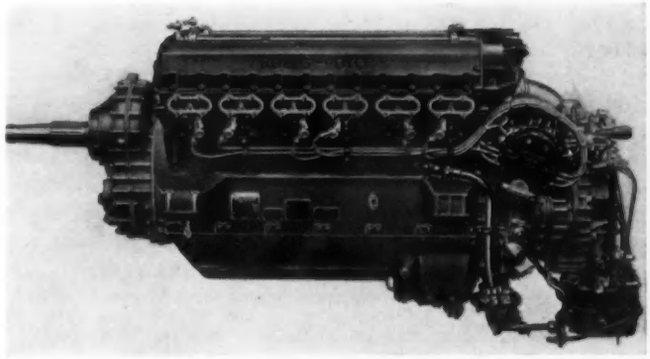


Fig. 4—Kestrel Engine (side)

engine cowling and, on single-engined aircraft, permits large-diameter airscrews without excessive under-carriage height. All transverse loads from the gearing are eliminated by the method of transmitting the drive from the crankshaft. This arrangement has a flange on its forward end to which is fitted an internally toothed ring engaging with corresponding teeth on the end of a hollow shaft mounted on the reduction-gear housing. The other end of the hollow shaft is formed with splines which transmit the torque to a hollow pinion mounted on roller bearings. This pinion engages with a toothed ring bolted to a flange integral with the airscrew shaft which, in turn, is supported on a roller bearing and a ball-and-thrust bearing. Figs. 2, 3, and 4 show front, rear, and side views respectively of the Kestrel engine.

Development of Kestrel Engine

The progress made in the various types of Rolls-Royce liquid-cooled Kestrel engines since their inception in 1927 is shown diagrammatically in Figs. 5, 6 and 7 in which crankshaft speeds, compression ratio, servicing period, brake mean effective pressure, and weight per brake horsepower for normal sea-level power and maximum performance ratings, are shown plotted on a time basis.

In each case there is a general decrease in the weight/power ratio, which decrease results from the increase in rotational speeds made possible by the improvements in bearing metals and from the improvements in knock ratings of fuels permitting increased power output from a given piston displacement by means of higher compression ratios and higher boost pressures.

It will be observed that, in spite of the higher duty which the engines are now called upon to fulfill, the permissible running time between overhauls has increased to the limit of 500 hr. allowed by the British Air Ministry.

Ratings on Special Fuels

For special occasions when high duty is required for short periods, the Kestrel has been cleared by the Air Ministry for operation at the following ratings using the fuels specified as shown in Table 2.

The approval tests satisfactorily cleared at the preceding ratings include power curves, detonation tests, high-speed tests, and two hours' endurance at nine-tenths of the full-throttle sea-level power.

It is of interest to record here that the standard fully supercharged Kestrel engines were operated by the Royal Air Force for some years without automatic boost controls, and we feel certain that there must have been many occasions when circumstances compelled pilots to take more power out

of their engines than was covered by the normal type-test ratings. In spite of this service no abnormal signs of distress were observed when the engines were returned for their periodic overhauls.

'R' Engine.—This engine of 6-in. bore, 6.60-in. stroke, twelve cylinders, 60-deg. V; and total capacity of 36.7 liters, was developed for racing purposes. Its construction is essentially similar to the Kestrel and, by resort to a special supercharger and fuel, allied with a life which, although short, was of sufficient duration to win the Schneider Trophy, this engine gave an astonishing power curve, two points from which are quoted.

R.P.M.	B.Hp.	B.M.E.P., lb. per sq. in.	Absolute Boost Pressure, lb. per sq. in.	Hp. per liter
3000	2585	305	32.50	70.50
3400	2783	290	35.50	75.60

The fuel used was made up of 20 per cent petrol, 70 per cent benzol, 10 per cent methanol plus 4 cc. of lead; it gives a knock rating of 92 by the modified motor method. Photographs of this engine are shown in Figs. 8, 9, and 10.

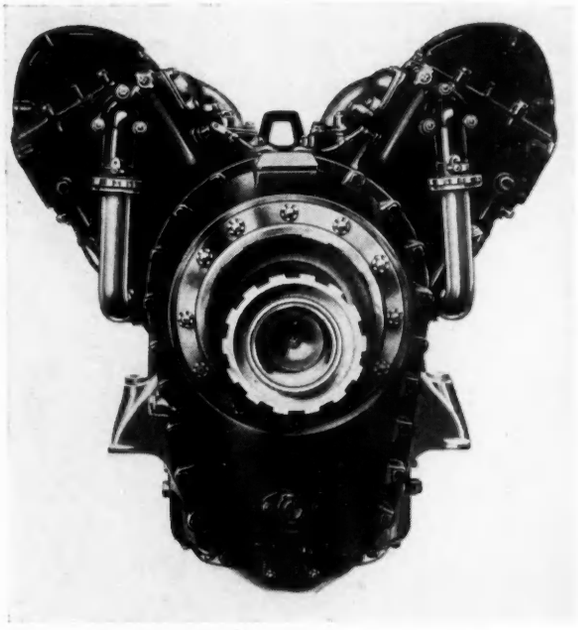


Fig. 8—Rolls-Royce 'R' Engine (front)

Merlin Engine

The Merlin is a fully supercharged engine designed for high-temperature liquid cooling. The cooling medium is ethylene glycol at normal and maximum outlet temperatures of 110 deg. cent. and 130 deg. cent. (230 and 266 deg. fahr.) respectively.

The bore and stroke are 5.40 in. and 6.00 in. respectively, giving a total engine capacity of 1647 cu. in. (27.0 liters). It is fitted with a reduction gear giving a ratio of 0.477:1 and a supercharger having a gear ratio of 8.59:1.

For operation on British Air Ministry D.T.D.230 fuel (87 octane), it is fitted with pistons giving a compression ratio of 6.0:1. This engine is in development and has been fitted and flown in new type aircraft.

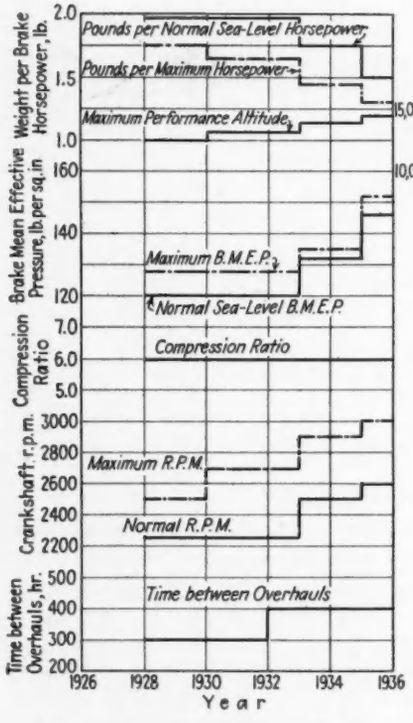


Fig. 5—Progress Diagram of Kestrel Liquid-Cooled Engines—Fully Supercharged Type

Bore, 5.0 in.; stroke, 5.50 in.; piston displacement, 1296 cu. in. No allowance for ramming effect of forward intake on maximum performance figures.

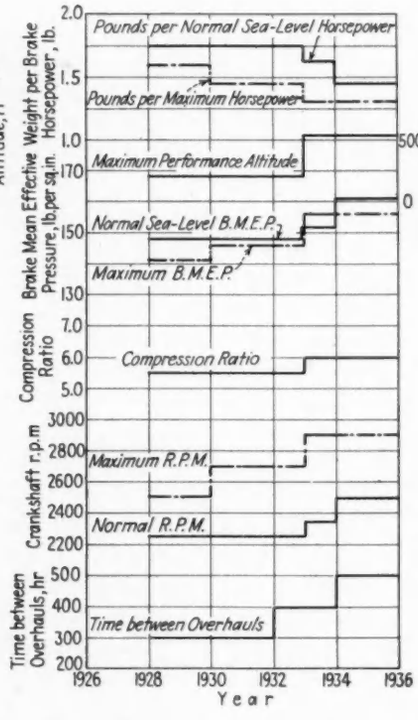


Fig. 6—Progress Diagram of Kestrel Liquid-Cooled Engines—Moderate Supercharged Type

Bore, 5.0 in.; stroke, 5.50 in.; piston displacement, 1296 cu. in.

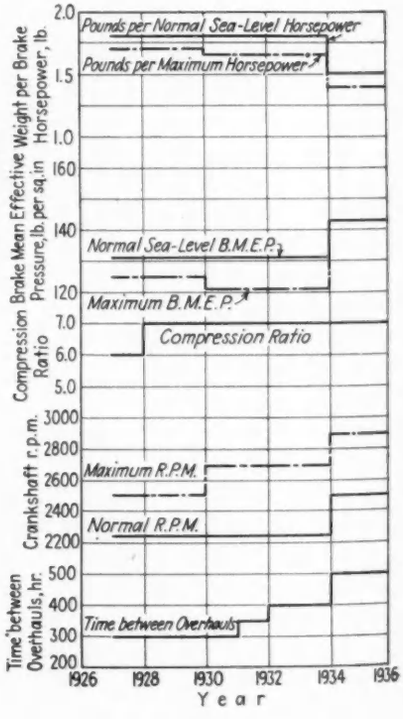


Fig. 7—Progress Diagram of Kestrel Liquid-Cooled Engines—Naturally-Aspirated Type

Bore, 5.0 in.; stroke, 5.50 in.; piston displacement, 1296 cu. in.

Table 2—Special Category Test Ratings—Full Throttle at Sea Level

Series	Normal Rating R.P.M.	Normal Rating B.Hp.	Normal Rating Altitude	Maximum Rating R.P.M.	Maximum Rating B.Hp.	Maximum Rating Altitude	Dry Weight, lb.	Compression Ratio	Boost at Normal Rating, lb. per sq. in.	Take-off Boost, lb. per sq. in.	Take-off R.P.M.
I-IIIs	2,250	710	Sea Level	2,700	835	Sea Level	920	6:1	+5.625	5.30	2,140
IV-VI	2,750	990	Sea Level	3,025	1,085	Sea Level	955	6:1	10.0	8.90	2,610

Series	Piston Speed, ft. per min. Normal	Piston Speed, ft. per min. Maximum	B.M.E.P., lb. per sq. in. Normal	B.M.E.P., lb. per sq. in. Maximum	Hp. per Liter Normal	Hp. per Liter Maximum	Fuel Consumption Specific Gravity	Fuel	R.P.M.	B.Hp.	Lb. per Hp-Hr.
I-IIIs	2,065	2,475	193	189.5	33.4	39.3	0.760	*D.T.D.134+5	2,250	710	0.508
IV-VI	2,525	2,770	220	219.5	46.5	51.2	0.843	†20/70/10+4	2,750	990	0.563

*D.T.D. 134 fuel +5 is 78 per cent petrol, 22 per cent benzol, 5 cc. tetraethyl lead per imperial gal., octane number approximately 90 by Air Ministry method.
†20/70/10 +4 fuel is 20 per cent petrol, 70 per cent benzol, 10 per cent methanol, 4 cc. tetraethyl lead per imperial gal., octane number approximately 92 by Air Ministry method.

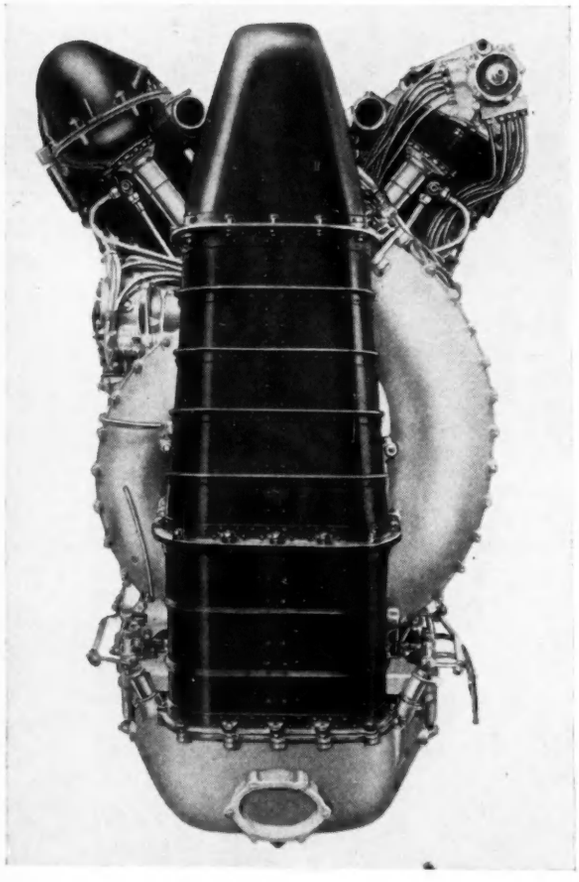


Fig. 9—Rolls-Royce 'R' Engine (rear)

The frontal area is shown in Fig. 1, and a front view of the engine is shown in Fig. 11.

High Performance Engines of the Future

Engine Problems.—Leaving what has been accomplished, views are advanced aimed at stimulating a discussion on the form the aero engine of the future will take to meet the demands of high-speed military aircraft using fuels of high knock rating. The outputs under consideration will be, it is estimated by 1940, at least 1600 b.hp. in a single power unit.

Table 3 is given imagining engines of this power operating at a piston speed of 3000 ft. per min. at 170 lb. per sq. in. m.e.p., but varying only the number of cylinders and estimating bore, stroke, revolutions per minute, and so on. For these engines the horsepower per square inch of piston area will be constant at 3.86 and the heat dissipation—based on 0.5 of the brake horsepower—also will be constant at 7.60 C.h.u.[1] per sq. in. per min.

For radial engines the overall diameter is approximately $6\frac{1}{2} \times$ bore $+ 2 \times$ stroke, and this value is taken for the diameter of all engines except the twelve cylinder one, which is assumed to be in V form, the diameter being $4 \times$ bore $+ 2 \times$ stroke.

The variation of bore, revolutions per minute, and horsepower per liter are plotted against the number of cylinders in Fig. 12.

Various cylinder arrangements are possible, such as single-, double- and triple-bank radials and V, broad-arrow and X dispositions on six-throw crankshafts. The principal gain to be expected from increasing the number of cylinders for a given power will be a reduction of the bulk of the engine and a decrease of the weight/power ratio.

It has been suggested that weight/power ratio cannot be reduced by increasing the number of cylinders as the metal thicknesses cannot be scaled down in proportion for reasons of casting, machining, or handling. This statement may be true for certain parts, but then we have the advantage of greater reliability or the capacity for development to higher powers.

The weight/power ratio should vary inversely as the square root of the number of cylinders, but such parts as reduction

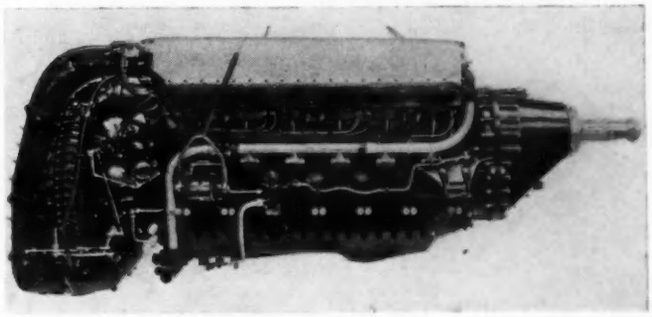


Fig. 10—Rolls-Royce 'R' Engine (side)

[1] Centigrade heat unit—the quantity of heat necessary to warm 1 lb. of pure water through 1 deg. cent., from 14 to 15 deg. cent., or approximately, 1 C.h.u. = 1.8 B.t.u.

Table 3—1600-Hp. Engines at 3000 Ft. per Min. and 170 Lb. per Sq. In. M.E.P.

Number of Cylinders	Square — Engine Bore = Stroke							Long Stroke = 1.2 Bore						
	Bore and Stroke, in.	R.P.M.	Capacity Cu. In.	Capacity Liters	Hp. per Cu. In.	Hp. per Liter	Diameter, in.	Stroke, in.	R.P.M.	Capacity Cu. In.	Capacity Liters	Hp. per Cu. In.	Hp. per Liter	Diameter, in.
9	7.64	2,360	3,210	52.7	.500	30.4	65	9.16	1,970	3,840	63.0	.417	25.4	68
12	6.64	2,720	2,800	45.8	.572	35.0	40	7.95	2,270	3,330	54.7	.480	29.2	43
14	6.12	2,950	2,570	42.2	.624	37.9	52	7.35	2,460	3,090	50.6	.518	31.6	55
16	5.73	3,140	2,420	39.7	.661	40.4	49	6.87	2,620	2,900	47.5	.553	33.7	51
18	5.40	3,340	2,270	37.2	.705	43.0	46	6.48	2,790	2,720	44.6	.590	35.6	48
21	5.0	3,600	2,060	33.8	.775	47.3	43	6.00	3,000	2,530	41.5	.624	38.6	45
24	4.68	3,850	1,970	32.3	.815	49.5	40	5.62	3,210	2,365	38.8	.676	41.3	42
30	4.18	4,300	1,760	28.9	.910	55.5	35	5.02	3,600	2,110	34.6	.760	46.2	37

gears and superchargers being independent of the cylinder size cannot be scaled down while the ignition system increases with the number of cylinders. The net result is that the weight/power ratio will be reduced in a lower proportion.

Another important feature of increasing the number of cylinders is the better detonation characteristics obtained because of decrease of cylinder size, increased speed of rotation, and better cooling. The decrease in cylinder size improves the detonation characteristics by shortening the flame travel. The area of combustion surface for absorbing heat is larger per unit volume and, at the same time, the heat-flow paths are shorter, both effects tending to improve the cooling and reduce local hot-spots.

These features are particularly valuable in their effects on the piston and poppet valves, which are now very critical members. Valve-seat distortion and cylinder-head cracking also should be reduced by the consequent increase in head stiffness. Other advantages are increased reliability and smoother running, although the 60-deg. V, twelve-cylinder engine is very satisfactory in these respects.

The disadvantages are, first and foremost, the expense of developing a different type of engine from the V.12 which

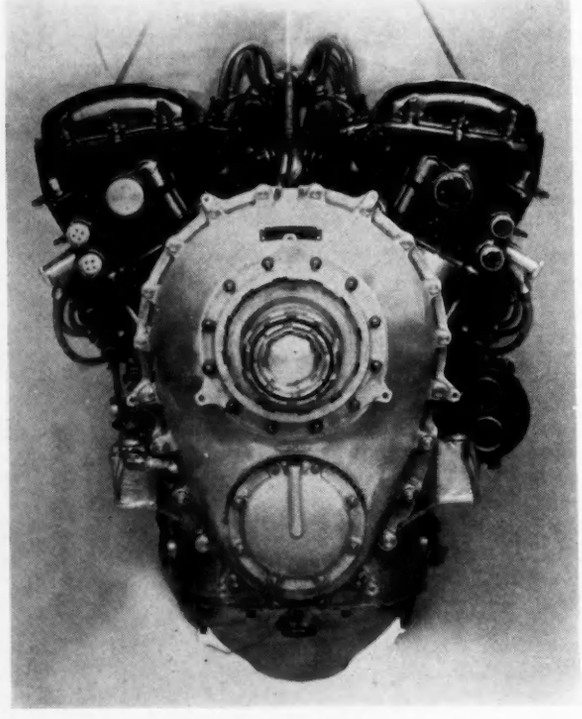


Fig. 11—Merlin Engine

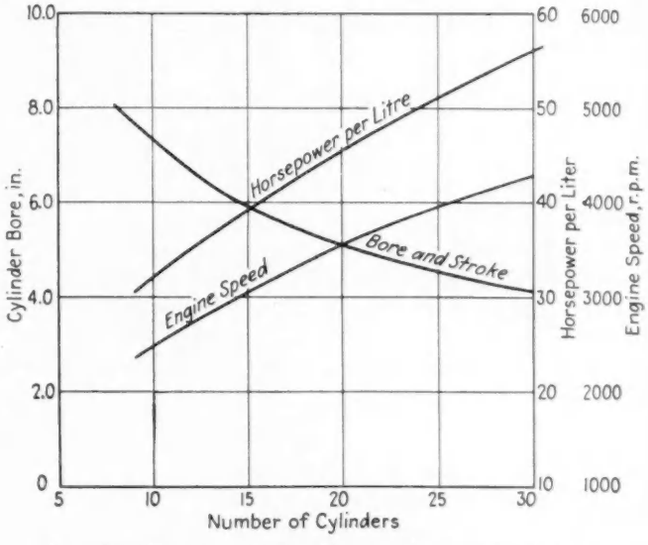


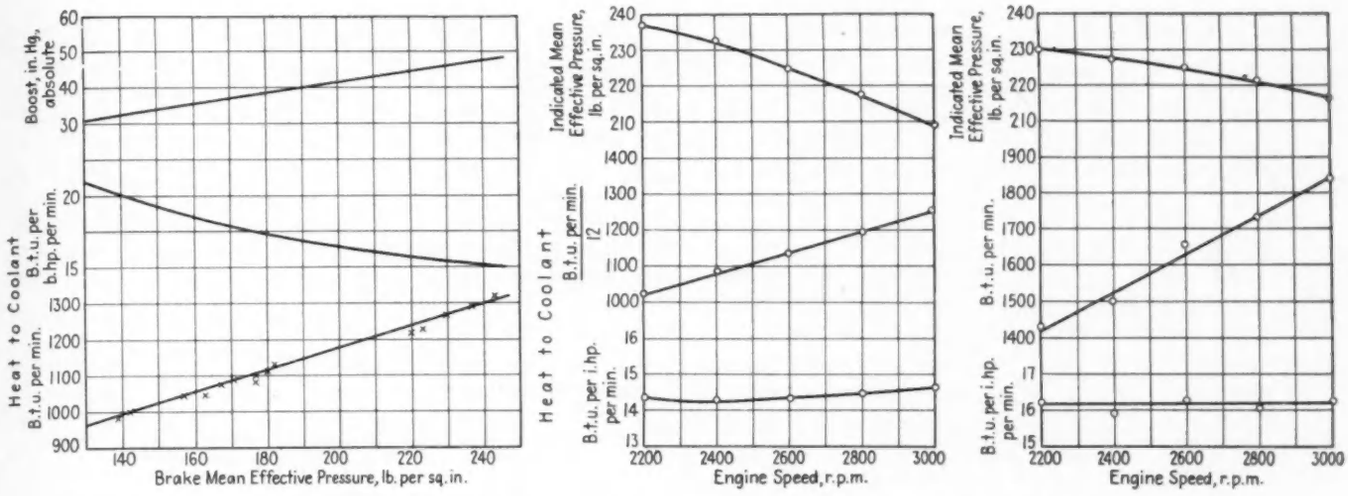
Fig. 12—Engine of the Future—Variation of Bore, Revolutions per Minute, and Horsepower per Liter Plotted against Number of Cylinders

Engine data for 1600 b.hp.: piston speed, 3000 ft. per min.; brake mean effective pressure, 170 lb. per sq. in.; brake horsepower per square inch piston area, 3.63; stroke/bore ratio, 1/1; heat dissipation, 7.60 C.h.u. per sq. in. per min.

has been in favor for the past 20 years. It may be argued that it is not necessary to depart from the V.12 when one considers that an engine of this type with 6- x 6.6-in. cylinders giving 2400 hp. on 92-octane fuel, won the Schneider Trophy in 1931. It should be remembered, however, that it was developed for one hour's operation only at this output. It should be emphasized, however, that there has never been any evidence that, even at the 2400-hp. output, we had reached the limit of heat dissipation. Secondly, more cylinders mean greater initial cost, and increased cost of maintenance and overhauls, as the amount of fitting required is increased.

If we consider the mechanical difficulties involved in increasing the number of cylinders, the first would appear to be the big-end and crankshaft assembly. By adding more rows to a radial or more banks to an in-line engine, we are faced with the question of solid big-ends and built-up crankshaft, or solid crankshaft and split big-ends. We prefer the latter arrangement, but we think there may be considerable difficulty in designing a split big-end to carry more than four rods. Then the necessity of operating at higher revolutions per minute involves higher bearing loads and considerably more difficulty in the design of a satisfactory valve gear.

The selection of engine form would appear to favor a


Boost, in. Hg, absolute
Heat to Coolant B.t.u. per b.hp. per min.
B.t.u. per min.
Brake Mean Effective Pressure, lb. per sq. in.
Indicated Mean Effective Pressure, lb. per sq. in.
Heat to Coolant B.t.u. per min. / 12
B.t.u. per i.hp. per min.
Engine Speed, r.p.m.
Indicated Mean Effective Pressure, lb. per sq. in.
B.t.u. per min.
B.t.u. per i.hp. per min.
Engine Speed, r.p.m.

Figs. 13, 13A, and 13B—Effect of Brake Mean Effective Pressure (Fig. 13) and Speed in Different Units (Figs. 13A and 13B) upon Heat-Flow to Coolant

(13) Single-cylinder units. R.p.m., 2600; bore, 5.0 in.; stroke, 5.50 in.; compression ratio, 5.85:1; fuel, D.T.D. 230; boost-air temperature, 70 deg. cent.; best ignition; weakest take-off mixture consumption; naturally aspirated motoring losses, 29 lb. per sq. in. m.e.p. Fabricated four-valve penthouse head. Evaporatively cooled; cylinder unlagged in still air. All pipe work lagged. No allowance for radiation losses.

(13A) Goshawk B-45. Boost, 6¾ lb. per sq. in.; fuel, D.T.D. 224 + 7 cc. tetraethyl lead per imperial gal.; mixture strength, weakest maintained maximum power; glycol, 110 deg. cent., outlet temperature; bore, 5.0 in.; stroke, 5.50 in.; compression ratio, 6.5:1.

(13B) Single-cylinder units. Bore, 5.40 in.; stroke, 6.0 in.; compression ratio, 6.05:1; boost air, 100 deg. cent., 6¾ lb. per sq. in.; weakest maintained maximum power mixture strength; fuel, D.T.D. 224 + 7 cc. tetraethyl lead per imperial gal.; four-valve ramp head; evaporatively cooled.

crankshaft having six throws with multi-pistons on each throw. Increasing the number of cranks reduces the frontal area of the engine, but the length does not increase proportionally because the fewer banks of cylinders give space between for fitting accessories. We believe that the radial engine will develop by increasing the number of crankpins and by reducing the number of cylinders per crank, and that the in-line engine will increase the number of cylinders per crank. Each type, therefore, will approach the other in regard to mechanical problems and general layout, and the differences in frontal area will diminish. But, for the case under consideration, the engine having the greatest number of crankpins will have the smallest frontal area. This predicted development will modify mechanical efficiencies, the tendency being to decrease on the radials and to increase on the in-line engines. Specific weights also will change more favorably in the case of the in-line engine.

The total waste heat is affected in two ways: that transferred by convection is increased owing to the smaller cylinder diameter, while the radiation is reduced. It would appear that these two factors balance one another and the waste heat remains the same, although the lower metal temperatures due to better cooling with the small cylinder tend to increase the heat flow slightly.

The best stroke-bore ratio is a matter of compromise. Shortening the stroke will tend to reduce the weight of a given engine, while the revolutions per minute and horsepower per liter are increased, but bearing loads, being proportional to the square of the speed, will increase. As bearings and valve gear are now critical points, it is difficult to reduce the stroke-bore ratio very much, although the sleeve-valve engine offers scope in this direction. Even with sleeve valves there may be difficulties with the porting to achieve high volumetric efficiency; while high compression ratios, which may be expected in the future, add further difficulties to the short-stroke engine.

To sum up, the advantages of a short-stroke engine are reduced frontal area, reduced weight, and a stiffer crankshaft; the disadvantages are higher bearing and valve-gear stresses and a flat compression space.

Waste Heat

The brake horsepower of an engine is proportional to the product of swept volume, revolutions per minute, and brake mean effective pressure. The maximum brake mean effective pressure that can be maintained will, in general, be greatest with small cylinders because to achieve this purpose demands a high thermal efficiency and a large amount of heat generated per unit volume of the cylinder. The heat generated will be proportional to the weight of the fuel/air mixture so that a high charge weight per cycle is demanded which, in turn, can best be achieved by increasing the degree of supercharge. The thermal efficiency is a function of the expansion ratio employed, which again can generally be made higher in smaller cylinders.

In the past it has been sufficient for supercharged engines to maintain normal ground-level pressure at the supercharged height, but lately the demand for greater aircraft performance has led to induction-pipe pressures considerably in excess of the 14.7 lb. per sq. in. value. This tendency will continue within the range of fuels available. There is sufficient published evidence to show that the proportion of waste heat to be dissipated decreases as the amount of boost increases, but the total quantity to be dissipated will increase with power. A typical curve illustrating these effects is shown in Fig. 13. We are faced then with the problem of the best form of cylinder to deal with the increase in waste heat inevitably following an increase in cylinder performance.

Considering existing types we find that data are available which are of material use in this connection. Assuming that all the heat is dissipated from the surface of a cylinder swept by piston (including the two ends but ignoring the surface of the compression space because this factor is variable between the types of engine) and that the heat to be dissipated is 0.5 of the brake horsepower, then at current ratings the heat dissipated per square inch of this surface per minute is

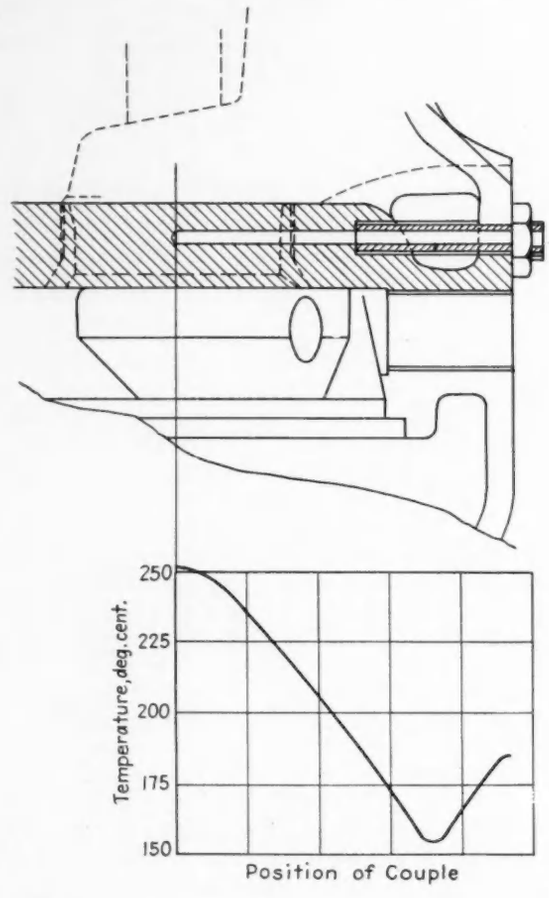


Fig. 14—Temperature Gradients Recorded by Withdrawing a Thermocouple along the Hole Drilled Between Two Exhaust Seats of Flat Four-Valve Kestrel Cylinder—Single-Cylinder Units

Glycol, 130 deg. cent. outlet temperature; b.m.e.p, 210 lb. per sq. in.; compression ratio, 6.2:1; 0.59 pt. per b.hp-hr. fuel consumption; r.p.m., 2400.

of the order of 6 C.h.u. for liquid-cooled and air-cooled engines.

We have no authentic information regarding special performances on the air-cooled engines but, as mentioned earlier, the Kestrel full supercharged engine has been cleared satisfactorily for flight duties in which the heat dissipation reaches 8 C.h.u. and, in the case of the 'R' engine that won the Schneider Trophy in 1931 at an output of 70 b.hp. per liter, the heat dissipation reached the figure of 12.50 C.h.u. When this engine was stripped after the race, the pistons were in perfect condition with no traces of distress from overheating. Ricardo has run a small liquid-cooled, sleeve-valve cylinder at an output of 107 b.hp. per liter corresponding to a heat dissipation of 10.50 C.h.u., and this engine ran satisfactorily for 100 hr. under these conditions[2]. One other case can be quoted and that is of a 24-cylinder air-cooled engine of high rotational speed and small cylinders at 45 b.hp. per liter in which the heat dissipation is 7.20 C.h.u.

These examples show that, associated with the large horsepower per square inch of piston area of the 'R' engine, is a very high specific-heat dissipation which was met satisfactorily by careful attention to the water circulation. The other liquid-cooled case, although having 50 per cent higher specific output, actually resulted in a lower specific-heat flow due to the beneficial effects of high rotational speed and high mean effective pressure and this result is substantiated further by the air-cooled case. Both cases illustrate also the ability of small cylinders to deal with high heat flows.

The first problem to be tackled in the disposal of waste heat is to limit as far as possible the heat actually transferred to the cylinder walls. A reduction of heat flow may be obtained in the following ways:

(1) By keeping the heated surface as small as possible, making the combustion space compact and having short and smooth exhaust ports.

(2) By increasing the compression ratio which increases the brake horsepower while reducing the percentage of waste heat.

(3) By operating with cylinder temperatures as high as practicable.

(4) By reducing the turbulence in the cylinder to no more than is sufficient for good combustion.

(5) By working below the detonation point of the fuel which, in turn, is affected by supercharger efficiency, induction systems, and form of cylinder-head, as well as by the knock rating of the fuel and by its mixture strength.

Owing to the limit of heat dissipation from a given fin-base area, air-cooled engines have had to be developed as two-valve or as penthouse types. This development has been facilitated by the fact that the outside surface temperature, which can be measured readily, is a direct indication of the rate of heat flow through the cylinder walls.

With liquid-cooled engines, the cooling properties are so very good inherently that the average metal temperature is readily kept down in spite of the severe temperature gradients brought on by defective circulation and varying wall thicknesses. These temperature gradients, however, lead to distortion and cracking of cast cylinder-heads.

With the substitution of glycol for water in the liquid-cooled engine the metal temperatures increase for three reasons: (1) Higher coolant temperatures employed. (2) Relatively poor heat-transfer coefficient of glycol. (3) Reduction of the evaporative effect, which with water cooling acts as a safety valve on hot-spots.

The following table gives the relevant properties of ethylene glycol and water:

	Density at 100 deg. cent.	Viscosity at 100 deg. cent.	Boiling Point, deg. cent.	Specific Heat	Latent Heat at 100 deg. cent.	Freezing Point, deg. cent.	Flash Point, deg. cent.
Water	0.96	0.003	100	1.020	540	0	——
Glycol (97 per cent)	1.05	0.020	174	0.675	240	—17	124

The heat-transfer coefficient of water is roughly 4.70 times that of glycol.

It will be seen that the two properties favoring ethylene glycol as a coolant are its high-boiling and low-freezing points, but it is inferior to water in the other properties.

As a result distortion and cracking become more serious when a changeover is made to ethylene glycol, and it becomes necessary to reconsider coolant flows and velocities in the same way as the air-cooled engine manufacturer arranges fins and directs the cooling air. In other words, it is not sufficient only to cast any sort of convenient jacket around the cylinder. The simple method of fitting thermocouples over the surface is not available in the liquid-cooled engine, and a

[2] See "The Internal Combustion Engine", Vol. II, by D. R. Pye, University Press, Oxford.

complete investigation using drilled holes for thermocouples is very difficult. Partial investigation, however, has disclosed the fact that temperatures of the order of 300 deg. cent. may be attained in the head castings of glycol-cooled engines where circulation is deficient (See Fig. 14).

The heat flow from the gases to the engine surfaces has been investigated theoretically, and it appears that at an engine power of 3.2 hp. per sq. in. of piston surface and a 6:1 compression ratio, the heat flows are of the following order: 9 C.h.u. per sq. in. per min. to compression space and piston crown; and 3 C.h.u. per sq. in. per min. at center of barrel and in exhaust ports.

Assuming that the surface in contact with the coolant is at the coolant temperature, the temperature gradients may be obtained graphically when it is found that, with normal water-cooled engines, the temperature drop through the metal may vary between 20 and 100 deg. cent. at different parts of the cylinder.

The assumption that the outer surface is at the temperature of the coolant is approximately correct for water cooling at 90 to 95 deg. cent., but not with glycol cooling. The reason for this discrepancy was investigated on a test rig with heat flows of 6 to 12 C.h.u. per sq. in. per min.; it was found that, whereas with water the surface temperature would not rise beyond 15 deg. cent. above the boiling point, the metal temperature with glycol at 110 deg. cent. rose to 200 deg. cent. The results of these tests are given in Fig. 15; they indicate that, whereas with water cooling the latent-heat effect keeps down the temperature of the hot-spots, with glycol cooling—(due to lower latent heat and specific heat) —the temperatures of hot-spots may reach air-cooled head temperatures. It is essential, therefore, with glycol cooling that the flow should be controlled and have a fairly high and uniform velocity over the head, although it is not essential to use fins to increase the cooling surface.

Also, more attention should be paid to the temperature gradients in the metal itself and to the density of heat flow at the cooling surface owing to the relatively poor heat-transfer coefficient of glycol and the ever increasing heat flow as engines are developed to higher specific powers. The cooling surface may be actually in the cylinder wall itself in the form of small drilled or coned holes at critical points.

As indicated previously, the cooling system has had to be reconsidered for glycol cooling, and so far the following modifications have been found necessary:

With the higher coolant temperatures the piston clearances first were increased but, by stepping up the coolant velocity around the cylinder jackets and regulating the flow, it was found possible to restore clearances to the water-cooled figures with considerable benefit to piston-rings and oil consumption.

Then on the head the hot-spots were treated by increasing the coolant velocity and directing its flow, as well as by providing more cooling surface and easier temperature gradients at the plugs, valve seats, and valve guides, so that the metal temperatures are now very little higher than those attained on the water-cooled engines.

Summarized Comparison of Results

A summary of the differences in engine performance when running on water at 85 deg. cent. outlet temperature as compared with glycol at 110 deg. cent. outlet temperature is:

(a) The difference in boost pressure and power is negligible.

(b) We invariably record a small reduction in specific consumption on glycol.

(c) The heat flow to jackets is of the order of 44 per cent of the brake horsepower with glycol and 54 per cent with water.

(d) The heat to oil is 4 per cent of the brake horsepower with glycol and 2 per cent with water.

Evaporative Cooling

Considerable work has been done on the development of evaporative cooling both in the engine and in aircraft. After some trouble in the early stages of the engine work, due chiefly to lack of appreciation of the modified conditions at the circulating-pump inlet, satisfactory type tests were completed on the Kestrel engine.

Some experimental aircraft installations have been made and flown, but the degree of success is not quite so complete. On obsolescent machines of comparatively low wing loading, sufficient aircraft surface can be made available to achieve the ideal of dragless cooling. The performance of these aircraft is such, however, that the complete elimination of cooling drag results in a comparatively small improvement in aircraft speed.

In considering the application of this method of cooling to high-performance aircraft, it will be appreciated that the surface available for cooling is restricted considerably and frequently very difficult of access involving some complication in the return of the condensate to the system. It should be emphasized that there is practically no benefit to cooling drag if the steam is condensed in a conventional honeycomb radiator. In view of these restrictions we have concentrated on high-temperature cooling with conventional radiators efficiently cowled.

Cylinder Construction

The complexity of a modern high-performance liquid-cooled cast head renders circulating passages and wall thicknesses extremely difficult to control dimensionally, with the result that there is always a tendency to put in more metal

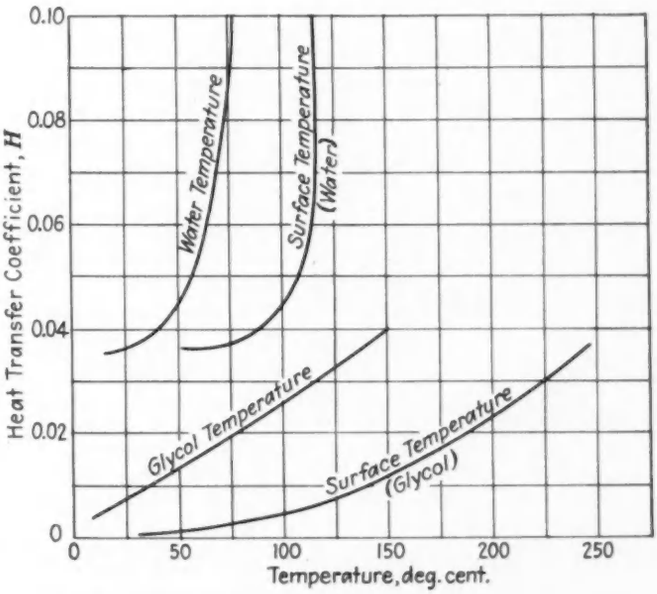


Fig. 15—Heat-Transfer Coefficients at 6 to 12 Centigrade Heat Units per Sq. In. Per Min.

Heat flow = $HV^{0.8}D^{0.2}$ C.h.u. per sq. in. per min. per deg. cent. Tests at 5 ft. per sec. in pipe diameter of 0.2 in.

than is necessary from stress considerations. Add to this difficulty the very small percentage of surface that can be machined, and it is not surprising that these heads are considerably heavier than they need be. This condition reacts unfavorably in other ways when operating temperatures as well as output are increased, the most serious defect being due to the temperature stresses set up by the complicated sections dealing with the heat flows. Finally, the materials available for making satisfactory castings of great complexity have very marked limitations in their mechanical properties, particularly at high temperatures.

We have, therefore, examined alternative methods of construction and have under consideration two methods, both of which have marked advantages over castings. The first and foremost advantage lies in the greater range of materials available. It is well known that the most successful British air-cooled engine has for some time had its cylinder-heads made from forgings in which modern methods of machining made possible by large production have enabled every square inch of surface to be machined, at a cost that compares very favorably with the cast head that it replaced. In addition, fully machined cooling surfaces and valve ports are what the designer intended with considerable benefit to cooling and valving respectively.

Our problem is more difficult in that an enclosure has to be formed to hold the coolant, but we are hopeful of finding some practicable solutions. One of the two methods has, in fact, been applied on earlier types of engine, but we think its utility is greatest in one particular type on which we are experimenting and, for which, the method solves several other difficult production problems.

Fuels and Detonation

This subject may be rather controversial, but we feel that it is necessary to ventilate all our results in order that the vexed problems of correlation of knock ratings and the phenomena of detonation may be considered from all angles.

The harmful effects of detonation are well known but, in the absence of any satisfactory direct means of measuring it in full-size cylinders, our investigations have been based on a constant intensity of audible knock confirmed periodically by destructive tests.

Table 4—Composition of Fuels

Bore 5.0 in. Kestrel Single Cylinder
Stroke 5.5 in. E.40042 Camshaft Boost-Air Temperature, 100 deg. cent. 896 B Plugs
R.P.M., 2600 Ignition Test

No.	Fuel	Specific Gravity	Per Cent Aromatics	Lead Content, cc. per imp.gal.	Octane No.	First Point B.M.E.P., lb. per sq. in.	First Point Pt. per B.Hp-Hr.	Last Point B.M.E.P., lb. per sq. in.	Remarks
	Compression Ratio 8.06 : 1 Evaporatively Cooled								
1	Iso-Octane Blend	0.723	Nil	4	96	140	0.470	232	56 Petrol 44 Octane Blended with Lead to give 98 Octane. Possibly due to deterioration in Stores this dropped to 96 Octane.
2	77 Octane + 7 (D.T.D. 224)	0.775	30	7	92	150	0.450	>248	Engine appeared rough.
3	50 Shell Av: + 50 Ben + 3	0.840	70	3	90	<134	Not known	>242	Engine appeared rough.
4	20 Pet/70 Ben/10 Methanol + 4	0.836	77	4	92	150	0.56	238	No detonation output limited by pre-ignition 238 lb. per sq. in. m.e.p. 8 per cent power drop by weakness at 0.83 pt. per hp-hr. No detonation. Cylinder blown off.
5	87 Octane Fuel (D.T.D. 230)	0.742	8	4	87	<130	....	156	
	Compression Ratio Dropped to 7 : 1								
6	77 Octane Fuel (D.T.D. 224)	0.757	30	Nil	77	<120	....	183	
7	77 Octane (D.T.D. 224)	0.716	Nil	Nil	77	<120	....	148	
8	87 Octane Fuel (D.T.D. 230)	0.742	8	4	87	131	0.496	200	
	Compression Ratio Dropped to 6.28 : 1 Coolant Changed to Glycol 150 deg. cent. Boost Air 90 deg. cent.								
9	77 Octane Fuel (D.T.D. 224)	0.757	30	Nil	77	<120	....	195	
10	77 Octane (D.T.D. 224)	0.716	Nil	Nil	77	<120	....	156	
	Compression Ratio Raised to 10.2 : 1 Evaporatively Cooled Boost Air 100 deg. cent.								
11	77 Octane + 7 (D.T.D. 224)	0.775	30	7	92	<130	....	>210	Detonation difficult to assess due to apparent piston rattle audible above the knock. This rattle affected plug voltage. Results at this ratio out of line with those at lower ratios.

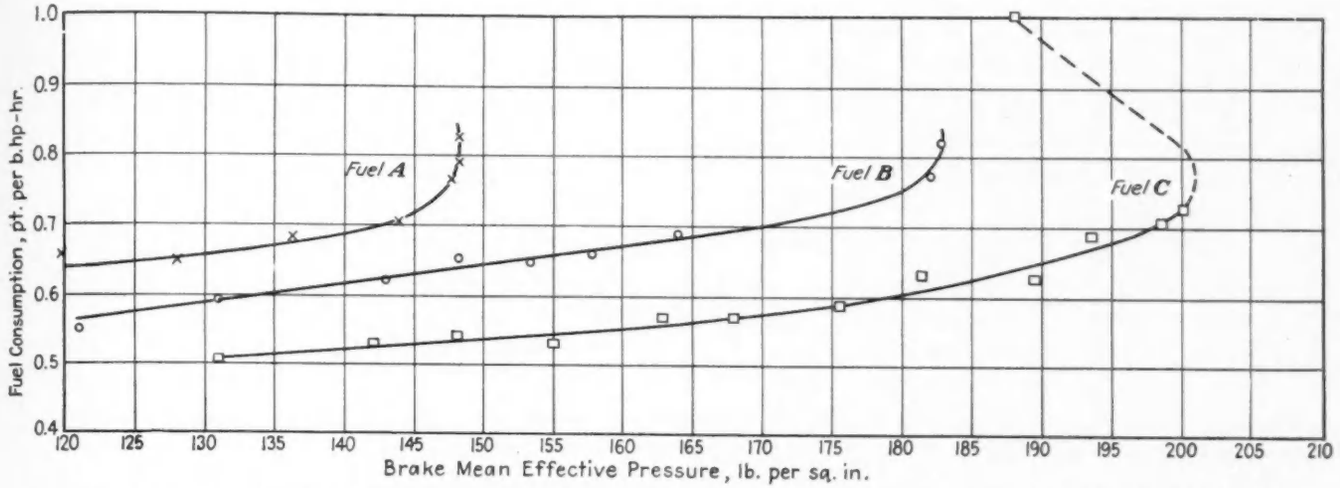


Fig. 16—Curves for Three Fuels—Increasing Brake Mean Effective Pressure for Constant Detonation—Single-Cylinder Units

Bore, 5.0 in.; stroke, 5.50 in; compression ratio, 7.1:1; four-valve flat head; evaporatively cooled; boost-air temperature, 100 deg. cent.; r.p.m., 2600.
Fuel *A*, Stanavo to D.T.D. 224, 77-octane, no aromatics, no lead, specific gravity, 0.716.
Fuel *B*, Anglo-American to D.T.D. 224, 77-octane, 30 per cent aromatics, no lead, specific gravity, 0.757.
Fuel *C*, Anglo-American to D.T.D. 230, 87-octane, 8 per cent aromatics, 4 cc. lead per imperial gal., specific gravity, 0.732.

There are three mixture strengths generally used for detonation tests, namely:

(a) At ½ per cent power drop due to weakness (now obsolete).

(b) At maximum economy consumption (cruising).

(c) At that consumption which will give maximum power for no detonation (maximum power and take-off).

The actual mixture strengths used in (c) will depend upon the combustion-chamber and the operating conditions. In some cases any increase in consumption above a certain point (on the rich side) actually increases the detonation, but normally any increase in consumption results in a reduction in detonation.

High-Octane Fuels

Some interesting results have been obtained with high-octane fuels with the Kestrel type of cylinder. Tests are as yet only in their early stages, but enough running has been done with single cylinders to show that the general trend is correct. This trend has been confirmed on the multicylinder engine.

These results are summarised in Table 4.

The "first point" is the maximum-economy condition, and the "last point" is the maximum-power condition.

It will be seen that the powers which can be carried bear little relation to the octane number (Air Ministry modified-motor method). As compared with Kestrel engine tests the C.F.R. engine very definitely derates aromatics and may derate high-lead-content fuels. Thus the 98/96-octane, non-aromatic iso-octane blend is not such a good fuel from the detonation standpoint as the 92-octane, 30 per cent aromatic D.T.D.224 + 7 cc. tetraethyl lead per imperial gal. Similarly the aromatic-free D.T.D.224 is 35 to 40 lb. per sq. in. m.e.p. worse than the normal D.T.D.224 petrol with 30 per cent aromatics. Further, the effect of adding 7 cc. of tetraethyl lead to the aromatic D.T.D.224 is to raise the end point from 183 lb. per sq. in. m.e.p. at 7:1 compression ratio to more than 210 lb. per sq. in. m.e.p. at 10.2 compression ratio and more, probably considerably more than 250 lb. per sq. in. m.e.p. at 8.06:1 compression ratio.

An apparent engine roughness was noticed with the high-aromatic fuels, and this roughness requires confirmation and further investigation before the performance with aromatic fuels can be correlated satisfactorily.

These results have been obtained over a very limited range of conditions (which nevertheless were representative of operating conditions), and it appears possible, at the very least, that the relative values of fuels will change with different engine conditions (that is, combustion-chamber design, operating temperatures and conditions). In fact, there is already a little evidence, in addition to the absence of correlation between the Kestrel and C.F.R. engines, that aero engines themselves will show serious lack of correlation.

Fig. 16 gives curves for three fuels of increasing mean effective pressure for a constant detonation intensity of 20 ft., comparing the behavior of two fuels blended to meet the D.T.D.224 specification (77-octane) and giving also the results of a fuel to the D.T.D.230 (87-octane) specification. It will be seen that the aromatic 224 fuel gives a much higher end-point than the non-aromatic, and this result is confirmed by Fig. 17, giving the behavior under take-off conditions of the same two fuels on an actual Kestrel engine of the type using this fuel in service.

Although we have not yet used 100-octane fuels on a Kestrel engine, we have fitted one with 7:1 compression ratio pistons and an improved induction system and taken consumption loops on D.T.D.230 (87-octane) fuel over a range of powers and speeds. These loops are shown plotted in Figs. 18 and 19 with some similar curves taken from the Wright Cyclone engine quoted in Raymond W. Young's paper read at the Annual Meeting of this Society, January, 1936[3]. It will be seen that, in spite of the lower knock rating of the fuel used in the Kestrel engine tests, the results compare very favorably with those obtained with the Cyclone engine and encourage us in the belief that the liquid-cooled engine has a great future on fuels of high knock ratings.

Engine Variables

The effects of various engine variables have been investigated only with D.T.D.230 (87-octane) fuel, and these effects may be considerably altered with other fuels.

Broadly, with D.T.D.230 fuel, the end points (maximum-power conditions) are affected by flame travel, speed, and

[3] See S.A.E. TRANSACTIONS, June, 1936, pp. 234-256; "Air-Cooled Radial Aircraft-Engine Performance Possibilities", by Raymond W. Young.

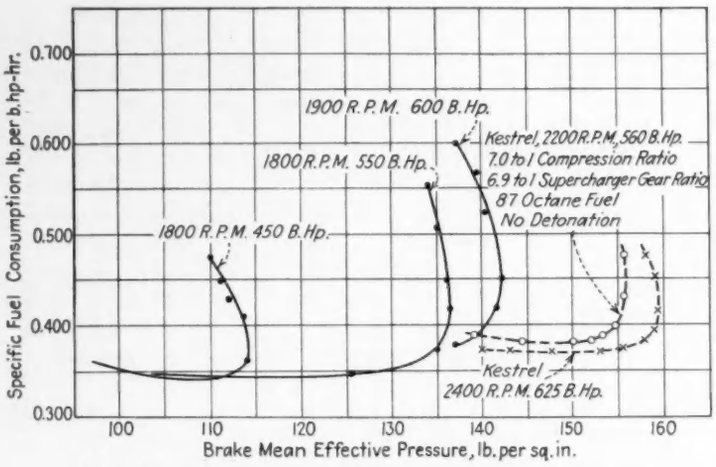


Fig. 18—Fuel-Consumption Loops at Various Speeds and Horsepowers—Rolls-Royce Kestrel Engine and Wright Cyclone R.1820G Engine

Wright Cyclone data: Compression ratio 7.85:1; supercharger gear ratio 5.95:1; 100-octane (U. S. Army Method) fuel.

1800 r.p.m. of Cyclone	2060 ft. per min. piston speed.
2200 r.p.m. of Kestrel	2020 ft. per min. piston speed.
1900 r.p.m. of Cyclone	2180 ft. per min. piston speed.
2400 r.p.m. of Kestrel	2200 ft. per min. piston speed.

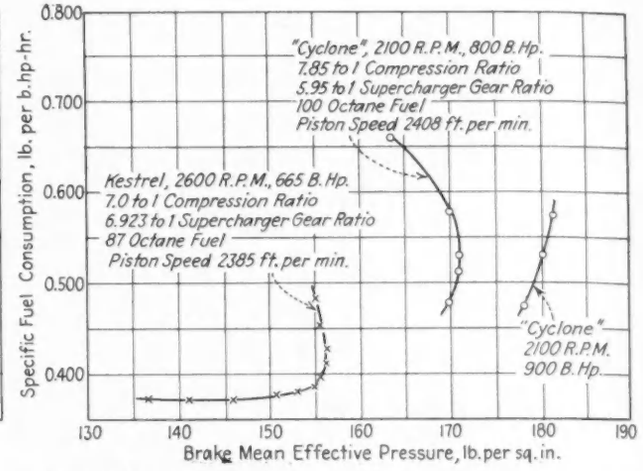


Fig. 19—Fuel-Consumption Loops at Various Speeds and Horsepowers—Rolls-Royce Kestrel Engine and Wright Cyclone R.1820G Engine

Comparative data	*Cyclone*	*Kestrel*
Bore, in.	6.125	5.000
Stroke, in.	6.875	5.500
Capacity, cu. in.	1823	1296
Supercharger rotor diameter, in.		8.500
Supercharger rotor tip width, in.		0.837
Rotor tip speed, ft. per sec.		668

compression ratio, whereas the first points (maximum-economy conditions) are affected in addition by head design and mixture and jacket temperatures.

Fig. 20 shows the effect of boost temperature on detonation at two compression ratios, and Fig. 21 shows the effect of jacket temperature. These curves illustrate the general trend, in which only the first points are susceptible to temperature. It would appear, therefore, that the air-cooled cylinder is at a very marked disadvantage for cruising power with its essentially higher operating temperatures.

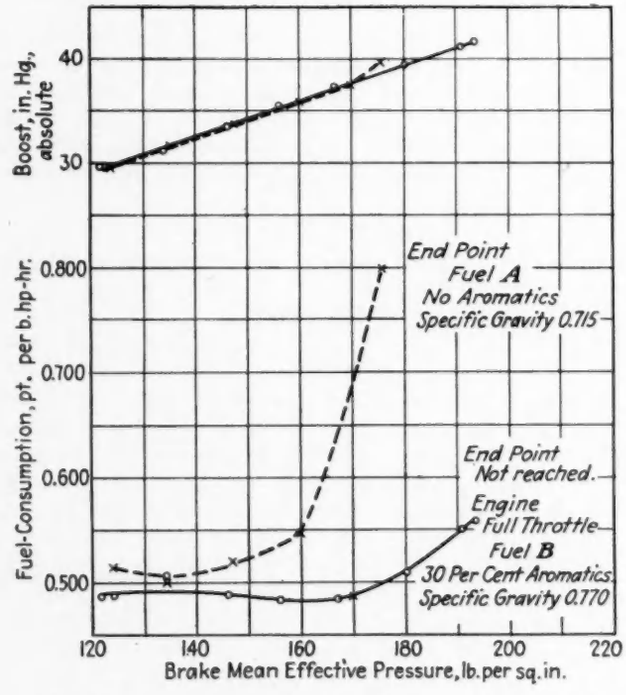


Fig. 17—Brake Mean Effective Pressure for Constant Detonation at Take-Off—Fuel with and without Aromatics

Kestrel V Engine 3727. Fuel comparison tests. Tests at 2150 r.p.m. (take-off). Water-cooled; 6.0:1 compression ratio; bore, 5.0 in.; stroke, 5.5 in.; 8.8:1 blower-gear ratio, four-valve flat head.
Fuel *A*, D.T.D. 224 with no aromatics.
Fuel *B*, D.T.D. 224 with 30 per cent aromatics.
Both fuels, 77-octane number and without lead.

In view of the fact that the maximum-power results are obtained at increasingly higher specific consumptions as the operating temperatures are increased, the constancy of the mean effective pressure could perhaps be explained by the greater internal cooling which the engine will permit at these higher operating temperatures. At the maximum-economy consumption, on the other hand, no such relief by internal cooling is available.

The effect of speed on detonation is shown in Fig. 22 with two opposed plugs and with one exhaust-plug only firing. As the speed is increased at a constant mean effective pressure, detonation will be reduced due to the shorter combustion time but increased because the greater power output makes the engine hotter. As the temperatures of the liquid-cooled engine are more nearly constant than are those of the air-cooled engine, it would be expected that air-cooled cylinders would show to disadvantage at high outputs. The ignition conditions can, however, mask this effect very considerably and this condition is brought out by the curves (Fig. 22) where, due to the reduced flame travel with two plugs, the first points lie parallel to but higher than those with one plug, whereas the end points with two plugs show, in addition, a rapid increase with speed. This last effect, which has been observed with several types of head, requires further investigation.

The effect of cylinder size has been ascertained and a scale-down in size of from 6.0-in. bore to 5.0-in. bore has enabled the head to be run at 10 per cent higher mean effective pressure. When a scale-down is made, the beneficial effects of reduced flame travel and combustion time are likely to be overshadowed by the changes in cooling efficiency that take place. For example, with a cast head, the smaller cylinders may make some of the water spaces so small that the cores become impracticable. Providing the coolant (liquid or air) passages can be scaled down, the smaller cylinder undoubtedly will score from a cooling standpoint. The heat paths—particularly from the piston—are shorter and the general reduction in metal thickness reduces temperature stresses,

enabling the unit not only to run at a higher mean effective pressure free from detonation but also to carry the consequent higher pressures.

Detonation Attack

When detonation occurs, thermocouples disposed around the inner surface of a Kestrel cylinder show rises in temperature of not more than 20 deg. cent., except in very extreme cases. Tests with cathode-ray engine indicators show no extreme pressure due to detonation, but only changes in the rate of pressure rise; it would seem, therefore, that detonation attack can be due neither to temperature nor to pressure. We have, therefore, investigated other possible causes for the attack and have succeeded in reproducing very similar effects in the chemical laboratory. The process also gives the same differentiation between the resistance to the attack of forged and cast materials as we experience in an engine under severe conditions of detonation, and we conclude that our investigations are on the right lines. Allied with this investigation and guided by its results, is an examination of methods of surface treatment to resist attack and an exploration of alternative materials having inherent lower susceptibility to the attack.

A point here in favor of the liquid-cooled engine is that it can run detonating for considerably longer periods than can the air-cooled unit because its temperature stability reduces the tendency of the increased head temperatures normally consequent on detonation to increase the detonation which, if allowed to take charge, sets up a vicious circle.

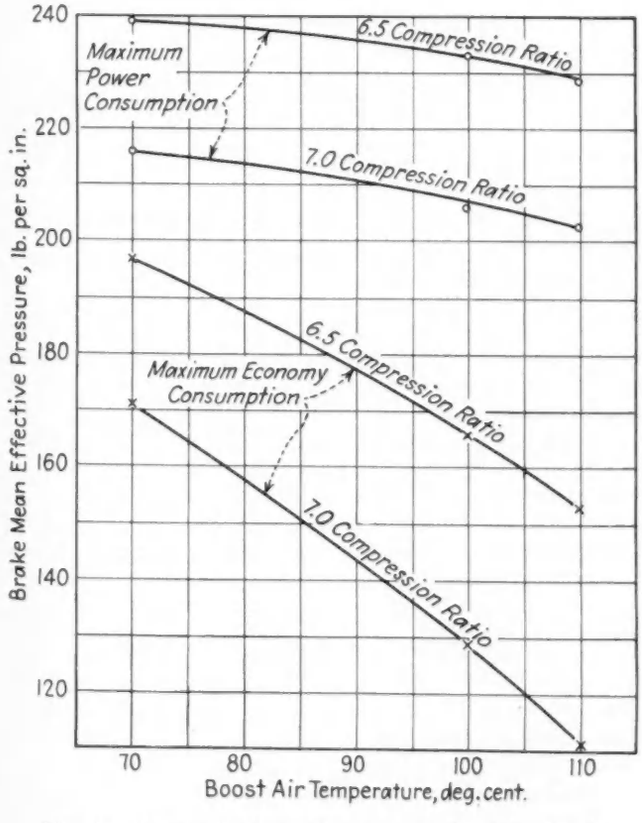


Fig. 20—Effect of Boost Pressure on Limiting Brake Mean Effective Pressure for No Detonation at Two Compression Ratios—Single-Cylinder Units

Bore, 5.65 in.; four-valve penthouse head; stroke, 6.00 in.; r.p.m., 3000; unboosted motoring losses, 30 lb. per sq. in. m.e.p.; 87-octane fuel (8 per cent aromatics).

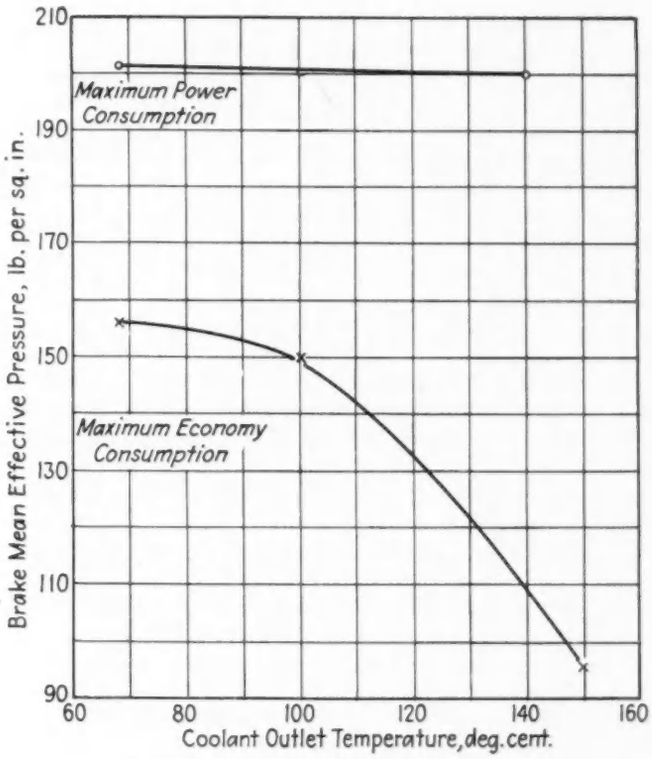


Fig. 21—Effect of Coolant Temperature Upon Limiting Brake Mean Effective Pressure for No Detonation—Single-Cylinder Units

Bore, 5.40 in.; four-valve turbulent head; stroke, 6.0 in.; boost-air temperature, 100 deg. cent.; compression ratio, 6.77:1; r.p.m., 2850; 87-octane fuel (8 per cent aromatics).

Single-Cylinder and Full-Scale Engines

Fig. 20 shows that, on a single cylinder, the end point is affected very little by increase in boost-air temperature. Fig. 23 is added, giving the average of a large number of tests on various main engines which shows a decrease in brake mean effective pressure at the end point as the mixture temperature increases.

The essential difference between the two cases lies in the method of carburetion which, for the single-cylinder unit, occurs after the blower and very close to the induction port; whereas, in the complete engine, it takes place before the blower and the mixture is conveyed from the blower discharge by a circular pipe to a distributing center, from which location manifolds deliver it to the individual cylinder ports.

We think that the explanation for the different temperature-detonation characteristics lies in the differing efficiencies of the petrol evaporation of the two cases. Preliminary tests on a single-cylinder unit employing both methods of carburetion confirm this explanation and have disclosed other features of the "carburetor-after-the-blower" system which require further investigation on a complete engine.

SECTION 2

Engine-Aircraft Problems

The chief aircraft problem is the dissipation of the waste heat for the minimum loss. The two cases under consideration, namely, liquid cooling and air cooling, present two aspects of the same problem, one indirect cooling and the other direct cooling. We believe that, by transferring the waste heat to a medium enveloping the cylinders which can then be

passed to a cooler designed for cooling only and disposed in most advantageous position on the aircraft, the necessary cooling can be achieved for less power than the direct air-cooled case where the heat must be dissipated at its source.

In addition, most liquids suitable for cooling have two very valuable automatic features which operate whenever engine conditions get very severe—such as are caused by improper use of a boost control or when short bursts of high power are required for difficult take-offs or when the engine is opened up after a long glide. I refer to the extra reservoir of heat capacity of the coolant over and above that of the engine itself and to its latent heat of evaporation. The former feature enables more heat to be absorbed under adverse conditions, or alternatively when power is low and cooling is high it forms a reservoir tending to maintain the engine surfaces at a uniform temperature, thus facilitating demands on the engine on opening up. The latter feature, namely, the latent-heat effect, comes into operation as a powerful addition to the sensible heat effect and enables head temperature to be kept low under exceptional conditions of heat dissipation.

For any given capacity engine there is a limit to the direct cooling surface that can be disposed on the cylinders. There is also a limit to the practical temperature at which the available materials can be operated and, therefore, to the mean temperature difference available for cooling; it follows, therefore, that there is also a limit to the amount of waste heat which can be dissipated. There is no such limit to the cooling of the same engine when liquid-cooled because the cooler, being independent of the engine, can be made to suit the power and, for the engine, the rate of circulation can be increased to suit the greater heat flows for a fractional increase in the proportion of the brake horsepower necessary to drive the circulation pump.

Our concern, therefore, is the extent to which the preceding limitations apply; whether, on the waste-heat side, we are yet in sight of the limit; and finally to examine for both cases the power necessary to be expended in order to dissipate the waste heat.

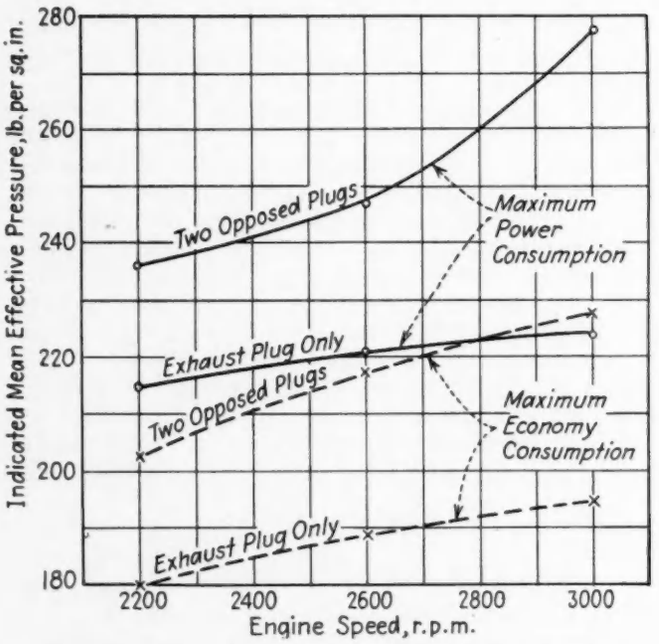


Fig. 22—Effect of Speed Upon Limiting Mean Effective Pressure for No Detonation—Single-Cylinder Units

Bore, 5.40 in.; four-valve ramp head; stroke, 6.0 in.; glycol-cooled; 110 deg. cent. outlet temperature; compression ratio, 6.54:1; boost-air temperature, 70 deg. cent.; 87-octane fuel (8 per cent aromatics).

Cooling for Highly Boosted Engines

When considering the cooling of highly boosted engines and the percentage of the engine power absorbed in cooling, it is preferable to consider it on a basis of fan cooling since cooling that is induced by the forward speed of the aircraft probably will prove impracticable for engines that develop full power at very high altitudes. Using a fan for producing the cooling flow through the radiator or engine, it should be possible to obtain the cooling for very little more power than that represented by the pressure drop of the stream through the cooling surfaces.

If the area of the cooling surfaces is fixed and if the condition for cooling is that the base-metal temperature of the engine remains constant, then the amount of cooling obtained will vary as $(\rho v)^{0.8} \times T$ for a tube radiator, and from $(\rho v)^{0.65} T$ to $(\rho v)^{0.5} T$ for an air-cooled engine, the actual index depending on the depth of the fins and their conductivity, where:

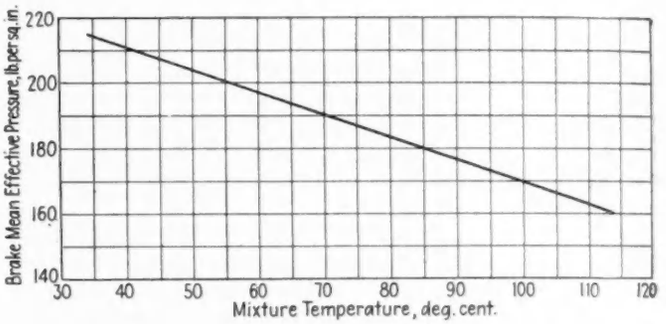


Fig. 23—Effect of Mixture Temperature upon Limiting Brake Mean Effective Pressure at Optimum Mixture Strength—Average of Tests on Various Rolls-Royce Engines

ρ is the air density.

v is the velocity in the cooling duct.

T is the available temperature difference between the engine and the cooling air.

For the high-temperature, liquid-cooled engine the product $\rho^{1/2} T$ is approximately constant with altitude, but this product decreases with altitude for the air-cooled engine. Depending on the actual base-metal temperature, the cooling power required for an air-cooled engine of constant power supercharged to various heights may decrease slightly or increase up to an index of $\rho^{-1.0}$ with altitude. For a liquid-cooled engine the cooling power required will increase slightly with altitude, but this increase will be very small, of the order of $\rho^{-0.1}$, and can be offset partly by the use of a slightly larger radiator employing a lower cooling-air velocity.

Varying Power—Constant Altitude

In most practical cases it can be assumed that the size of the liquid-cooled radiator can be increased proportionately to the engine power, in which case the ratio of cooling power to engine power will be constant, except in so far as this ratio is reduced by the reduction in specific heat flow to the coolant normally accompanying an increase in mean effective pressure.

In the case of air-cooled engines, however, having fixed cooling surfaces, the amount of cooling varies from $(\rho v)^{0.65} T$ to $(\rho v)^{0.5} T$ depending on the efficiency of the fins, and this product must be proportional to H the engine power.

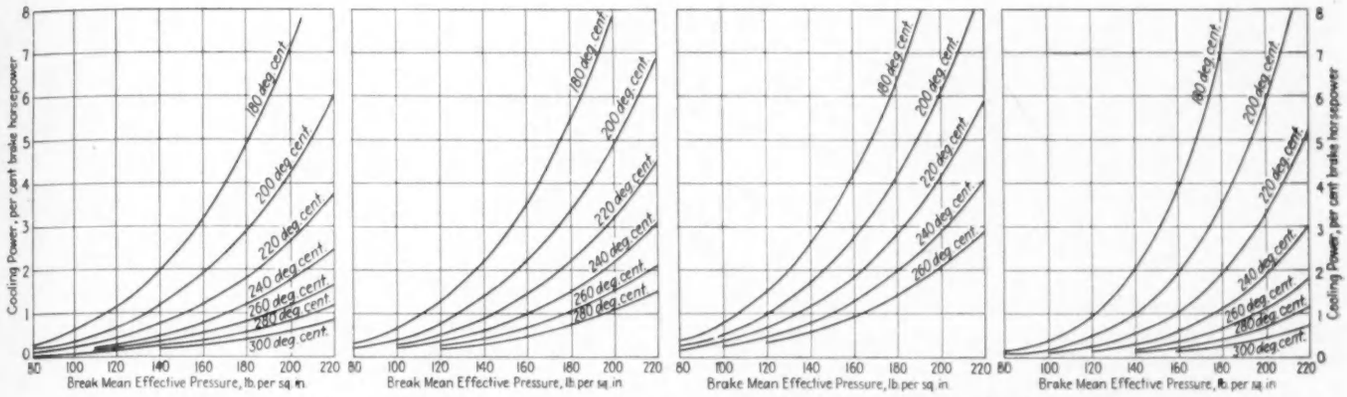

Cooling Power, per cent brake horsepower
180 deg. cent.
200 deg. cent.
220 deg. cent.
240 deg. cent.
260 deg. cent.
280 deg. cent.
300 deg. cent.
Break Mean Effective Pressure, lb. per sq. in.
Brake Mean Effective Pressure, lb. per sq. in.

Fig. 24—Cooling-Power Curves for Various Fin-Base Temperatures Plotted Against Brake Mean Effective Pressure

24-A—Ground level, estimated from $\left(\frac{H}{T}\right)^{4.6}$ 24-B—Altitude, 10,000 ft., estimated from $\left(\frac{H}{T}\right)^{4.6}$ 24-C—Altitude, 20,000 ft., estimated from $\left(\frac{H}{T}\right)^{4.6}$ 24-D—Ground level, estimated from $\left(\frac{H}{T}\right)^{6}$

Then H varies as $(\rho v)^{0.65}T$, *or* v varies as

$$\left(\frac{H}{T}\right)^{1.54} \times \frac{1}{\rho}.$$

Cooling drag power D varies as ρv^3, that is, as

$\left(\frac{H}{T}\right)^{4.6}\frac{1}{\rho^2}$ for the one case and as

$\left(\frac{H}{T}\right)^{6}\frac{1}{\rho^2}$ for the other case.

Thus, increase in power from a given cylinder, that is, an increase in H, is accompanied by a prohibitive rise in cooling drag unless T also can be increased. Hence, unless improved fuels, lubricating oils, and metals are made available enabling marked increases in the permissible value of T to be realized, the maximum powers at which air-cooled engines can be run are strictly limited by cooling-power considerations. Fig. 24 gives curves illustrating the foregoing effects. Small improvements can be obtained by using smaller size fins on high-boost engines, the pitch and depth of the fins being scaled down equally, but even in the limiting case with constant-temperature fins the cooling power will vary as $H^{3.75}$.

As an example of the cooling power necessary for an air-cooled engine at 140 lb. per sq. in. m.e.p. at ground level, the loss of pressure of the air passing through the cylinder fins corresponds to about 2 per cent of the engine horsepower. At 200 lb. per sq. in. m.e.p. the ratio of the cooling power to engine power would be increased by from 3½ to 6 times, that is, to somewhere between 7 and 12 per cent of the engine power. Except for a racing type of aircraft it would be quite impracticable to attempt to induce the required cooling flow by the forward speed of the aircraft, a separate cooling fan being almost essential, but, with a liquid-cooling engine, the radiator size can be increased as necessary and the cooling power is about 1 per cent of the engine power either at 140 or 200 lb. per sq. in. m.e.p.

Induced Flow—Duct Cooling

Where the cooling air is not circulated by a fan but is induced entirely by the airplane or slipstream speed, the cost of cooling is much greater. Modern development has indicated that the cooling flow is better controlled by means of varying the exit from the duct rather than by using shutters at the front or inside of the duct. The total cost of the cooling is usually two or three times the actual useful loss of pressure through the cooling surfaces. This amount is due to two main causes:

(1) To the loss of pressure or energy in slowing up the air in front of the cooling surfaces. This loss is particularly bad in the case of radial air-cooled engines, and no efficient combination of spinner and cowl front has yet been developed that will slow down or expand the cooling stream without loss.

(2) To the external surface of the cooling duct increasing the total skin-friction drag of the airplane. This loss becomes important in the case of high-speed aircraft with cowled low-velocity radiators external to the fuselage.

There are also other terms that increase the total cost of cooling, such as the actuator-disc efficiency for the duct, which is similar to an inverse airscrew efficiency, and the effect of the cooling air-wake on the remainder of the fuselage.

The lowest figures yet obtained for the total cooling cost of an ethylene-glycol-cooled installation are of the order of 2½ per cent of engine b.hp. at 200 m.p.h. and 3½ per cent of engine b.hp. at 300 m.p.h.

For radial air-cooled engines, even with fully controlled cowling operating at 300 m.p.h., we would predict from the foregoing data a considerably higher figure for the cooling power, which figure also will increase rapidly with the mean effective pressure of the engine, resulting in rather more than 15 per cent of the brake horsepower at 200 lb. per sq. in. m.e.p. for 300 m.p.h.

The in-line type of air-cooled engine has good possibilities at moderate mean effective pressures up to about 160 lb. per sq. in.; it lends itself well to low-velocity duct cooling and should give results comparable with the high-temperature liquid-cooled installations.

Engine Cooling System

The weight and drag of cooling systems become of major importance when considered in relation to clean modern aircraft possessing a high power/weight ratio of powerplant.

With an uncowled free air-flow radiator using water at or near the boiling point as the cooling medium, the necessary size is determined by the full-throttle climbing conditions, the air velocity over the cooling surface and the heat dissipation being approximately proportional to the machine speed, whilst the power absorbed in overcoming its drag increases as the cube of the speed.

It has been demonstrated, both by model and full-scale flight tests, that a considerable reduction in drag can be effected by cowling the radiator and by regulating the flow of cooling air by means of an adjustable flap at the rear exit.

A theoretical investigation regarding the characteristics of

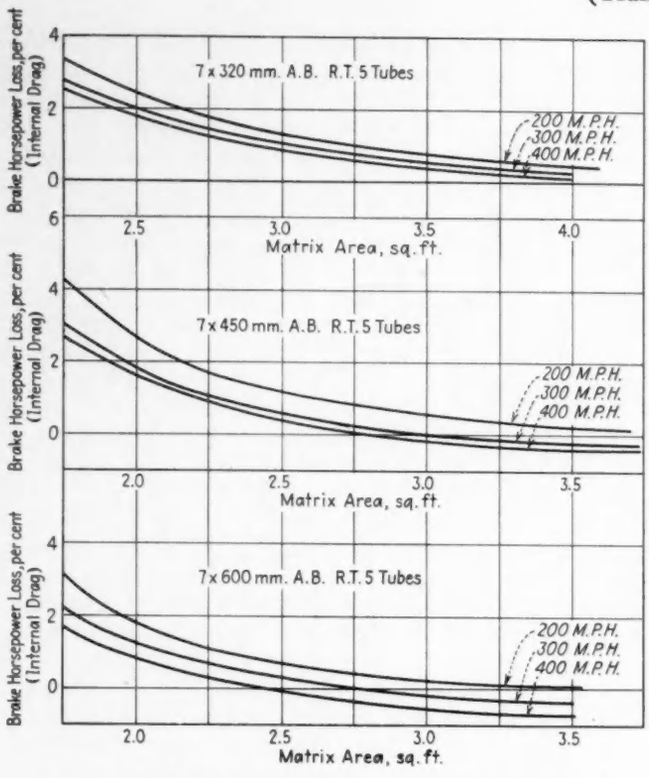

7 x 320 mm. A.B. R.T. 5 Tubes
200 M.P.H.
300 M.P.H.
400 M.P.H.
Brake Horsepower Loss, per cent (Internal Drag)
Matrix Area, sq. ft.
7 x 450 mm. A.B. R.T. 5 Tubes
200 M.P.H.
300 M.P.H.
400 M.P.H.
Brake Horsepower Loss, per cent (Internal Drag)
Matrix Area, sq. ft.
7 x 600 mm. A.B. R.T. 5 Tubes
200 M.P.H.
300 M.P.H.
400 M.P.H.
Brake Horsepower Loss, per cent (Internal Drag)
Matrix Area, sq. ft.

Fig. 25—Radiator-Drag Characteristics—Variation with Machine Speed for Different Tube Lengths

1000-b.hp. engine at 15,000 ft.; English summer conditions; level flight.

cowled radiators from a cooling, drag, and weight point of view has recently been carried out, the basic data as regards heat dissipation and drag characteristics of various types of radiator matrix having been determined by means of model tests. The investigation has embraced high-temperature cooling using ethylene glycol as the cooling medium and also the possibilities of operating sealed-water systems under varying degrees of pressure. Various types and lengths of radiator tube have been tried with the specific object of finding the best compromise for maximum cooling with minimum weight and drag.

The model tests gave figures of heat dissipation per unit of surface per degree difference in temperature between the mean liquid and the entering air, and also of the drag characteristics per unit of surface when considered under normal sea-level conditions of temperature and pressure for an entering-air velocity of 100 ft. per sec.

It was found that heat dissipation is approximately proportional to $(\rho v)^{0.83}$, where the symbols ρ and v apply to air density and air velocity at the faces of the matrix. Thus the effects of changes in air density and air velocity can be determined from the known sea-level conditions.

Similarly the pressure-drop characteristics with cold surfaces were found to be approximately proportional to $\rho v^{1.83}$ and thus, from the known sea-level conditions, the cold-surface pressure drop for any other conditions of density and air velocity can be found. It was found that some amplification is necessary when dealing with hot surfaces due to the change in density of the air passing through the tube, a condition that produces an increase in the drag characteristic. The model tests showed the increase over the cold-surface condition to be directly proportional to the rise in air temperature and inversely proportional to the air temperature at the radiator face, multiplied by a constant depending on the sectional area of the radiator tube. This factor should be taken into account, therefore, in dealing with radiator calculations.

It was found that, under the same conditions of tube length and air velocity, the heat-dissipation characteristics with water as the cooling medium were approximately 10 per cent higher than those with ethylene glycol; this percentage gain was obtained over a whole range of length/diameter ratios from 14/1 to 70/1. This effect arises from the poor heat-transfer properties of ethylene glycol as compared with water.

The temperature rise of the air passing through the radiator can be estimated, knowing the heat dissipated by the radiator and the weight of air passing through the tube block.

Applying the preceding method to cowled radiators, therefore, and assuming adiabatic compression of the air in the entry cowl up to the matrix face and adiabatic expansion in the exit cowl to atmospheric conditions, the pressure, temperature, and velocity conditions at all points within the cowling can be determined. The necessary cowl exit area for any desired coolant-temperature condition can be determined by equating the weight of air passing through the radiator to that passing through the cowl exit; and the internal drag, that is, the drag induced in producing the necessary cooling, can be calculated from the change in momentum of the air passing through the tube block.

Of all the practical matrices tried the most suitable as regards the best combination of cooling and low internal-drag characteristics was shown by the model tests to be the Anderton Brown honeycomb type.

To assess the true cost of a radiator in terms of engine power, however, other factors besides internal-drag loss have to be considered. The deflection of air around the radiator sets up considerable external drag, and it has been shown by tests of radiators in a free air stream that only 50 per cent of the approaching air actually passes through the matrix. The effect of cowling the radiator is to reduce greatly the external-drag losses by deflecting the major portion of the air around the outside of the radiator and reducing the velocity of the air passing through it. Additionally, the weight of the entire cooling system is responsible for further power loss which must be taken into consideration.

Internal-Drag Losses

The analysis shows that, for constant machine speed, the radiator surface required to perform a given amount of cooling decreases with increasing velocity at the face of the matrix, and that the internal-drag losses increase with increasing velocity. Thus, for any one speed condition, the larger the radiator the lower will be the internal drag. (See Fig. 25).

Internal-drag losses of tubes of the same type but of different sectional areas have virtually the same characteristics and magnitude, providing the length/diameter ratio remains unchanged and so, if for any reason there is a limitation on the permissible tube length, the tube of smaller cross-sectional area will require a lower frontal area to perform a given amount of cooling.

The effect of increasing the length/diameter ratio of tubes of the same cross sectional area, at any one machine speed, is to reduce the internal-drag losses. This statement means that with longer tubes smaller radiators can be used for the same internal drag.

Considering internal-drag losses, therefore, it appears that

high velocities in the matrix are inefficient since the increased power absorbed in overcoming the drag more than outweighs the increased heat dissipation.

External-Drag Losses

The external drag due to radiator cowling has been found by wind-tunnel tests to be approximately 1 lb. per sq. ft. of matrix area at ground density at an air velocity of 100 ft. per sec., from which figure the external drag for other density and velocity conditions can be estimated.

The external-drag power varies directly as the cube of the speed and as the radiator size; it increases very rapidly, therefore, with increases in machine speed.

Neglecting weight loss for the moment, since the aircraft manufacturer has to cater to the weight of the cooling system in his design, and considering only internal and external losses, it is seen from Fig. 26 that, as the radiator size is increased, the power loss decreases to a minimum and then increases again. A considerable reduction in the radiator size can be made by agreeing to a small increase in the minimum loss. This reduction in radiator size means a corresponding reduction in the weight losses. A further point is that, for tubes of the same cross-sectional area, the advantage as regards combined internal and external drag rests with the longer tubes which give minimum loss combined with minimum radiator area.

It would seem, therefore, that the logical step is to eliminate external drag entirely by enclosing the radiator in the fuselage or wings.

Power Loss Due to Weight

Taking the weight of the cooling system into consideration it is found that the length/diameter ratio for minimum loss increases with increasing machine speed, thus at 200 m.p.h. the best ratio would appear to be about 50/1; at 300 m.p.h., 65/1; and at 400 m.p.h., about 85/1.

Effect of Temperature

The effect of allowing the coolant temperature to increase at any machine speed is to reduce the radiator area necessary for a given amount of cooling and to reduce also the cooling drag of the radiator by an amount which is directly proportional to the mean temperature difference.

Reduction of Plumbing Troubles

It has been found possible to produce a very compact powerplant by mounting the radiator directly under the engine crankcase and by fitting a small header tank around the reduction gear at the nose. This arrangement means that the amount of pipe work in the installation is reduced to a minimum. Furthermore, a very simple and efficient type of pipe joint has been produced in which a special high-temperature-resisting rubber is employed. Radiators are made to withstand considerable internal pressure, and a high-temperature solder is used in their construction. The elimination of the retractable radiator in favor of the cowled type also has simplified the cooling system considerably since movable joints do not have to be provided.

The combination of these factors has produced a system in which plumbing troubles are virtually non-existent.

Oil Cooling and Exhaust System

In modern high-output engines operating at high jacket temperatures, the heat transference to oil is high and the question of oil cooling assumes considerable importance.

The same characteristics as those of the main cooling system are required, namely, maximum cooling for minimum drag. It is considered that a stage in the design of machines has now been reached where any spoiling of the machine lines by external oil coolers will produce a considerable drag. Machine surface types are ideal since they incur no additional drag but, in general, the area available is inadequate for dealing with the quantity of heat to be dissipated and their vulnerability for use in military aircraft renders them unsuitable.

The modern tendency, therefore, is to combine the radiator and oil coolers in the same tube block and locate them within the same pressure cowling. Additionally, the modern tendency towards single-entry up-draft carburetors involves location of the mouth of the air intake below the fuselage instead of at the sides, as located hitherto. In view of the troubles encountered with ice formation in carburetors and the necessity for providing some form of heated-air intake, the logical solution appears to be that of building the intake into the radiator matrix and surrounding it by the oil cooler. By this means the cleanest possible body lines are secured and the only excrescence on the fuselage is the cowling surrounding the oil cooler, radiator, and air intake.

The general requirements of an exhaust system are:

(1) Low drag and weight.
(2) Efficient flame damping.
(3) Good silencing properties.
(4) No interference with pilot's view.
(5) Good service life.

Flame damping and the suppression of noise are allied with the degree of cooling and gas expansion but, in order to meet the weight and drag conditions, the size and surface must be reduced to a low figure.

Until recently the type used in service consisted of a long-tail pipe extending from the manifold in which expansion and cooling were effected, but this type suffered from exces-

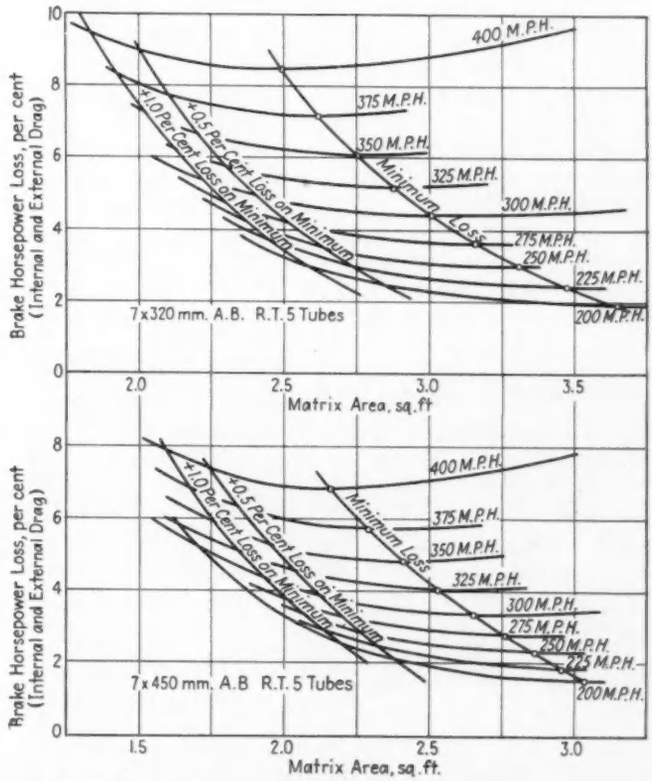


Fig. 26—Radiator- and Cowling-Drag Characteristics—Variation with Machine Speed for Two Different Tube Lengths

1000-b.hp. engine at 15,000 ft.; English summer conditions; level flight.

sive weight and drag and, in addition, to the adverse effect of length on engine power. Also, the resonant note produced had a nerve-racking effect on the pilot.

These types were replaced by what has been termed the "ram's-horn" type in which expansion takes place in one lobe coupled to each pair of cylinders. This manifold is shown in Fig. 27. Originally the lobes were perforated for gas exit, but these types produced flaming under certain conditions because the distribution of the hot gases over the surface of the lobe isolated the metal from the cooling air.

This difficulty was overcome by making the gas exit in the form of a slot on the underside of each lobe and, whereas each lobe was formerly streamlined independently, it has been found possible to reduce drag considerably by merging the lobes into one streamline contour. They are found to give a noise reduction of 6 decibels as compared with short stub pipes; they are approximately 40 per cent lighter than the long-tail-pipe types; they give a worth while reduction in drag; and the back pressure set up is negligible.

These manifolds were produced originally in low-carbon steel to British Air Ministry specification D.T.D.12.A. but, in view of our generally unsatisfactory experience with this material, stainless steel to specification D.T.D.171.A. was substituted and is now being used with satisfactory results.

Inconel, an alloy of nickel chromium and iron, is also being used experimentally. It is found to possess a very high heat and oxidation resistance.

Comparisons of Installed Weights

An investigation into the completely installed weights of liquid-cooled and air-cooled engines of virtually the same power has been made recently. Taking into account the weights of engine mounting, cowling, exhaust system, airscrew, coolant, and so on, it has been found that the difference in installed weight lies solely in the difference between the net dry weights of the two engines, the balance on current-type engines being in favor of the air-cooled type. This finding refutes the argument that the weight of an installed liquid-cooled engine is very much greater than that of an air-cooled engine of equal power.

We think when more experience has been obtained with variable-pitch airscrews that it will be found possible to take advantage of the smoother torque of the V.12 engine to use lighter hubs than are found necessary on the single-row radial engines.

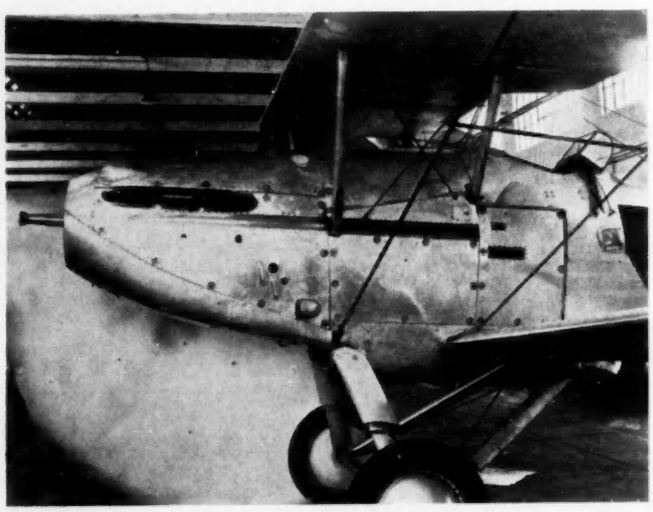


Fig. 27—"Ramshorn" Manifold

An analysis of the installation weights follows:

Comparative Estimates of Installation Weights Based on Engines of 1000-B.hp. Rating.

Component	Radial Air-Cooled, lb.	Liquid-Cooled, lb.
Airscrew	360	360
Cowling	100	60
Mounting	100	60
Exhaust Manifolds	120	35
Radiator	....	63
Header Tank	....	15
Piping	....	8
Coolant	....	115
Oil Cooler	25	10
Fuel Piping	20	20
Oil Piping	10	10
Tanks	110	110
TOTALS	845	866

Airscrew Characteristics

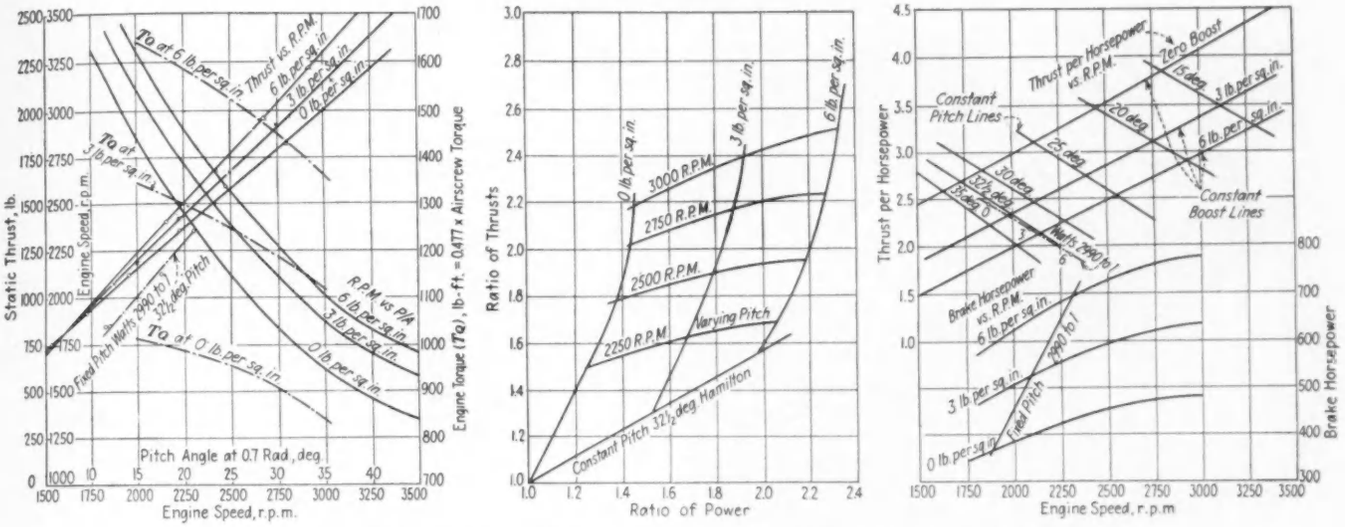
The increase in forward speeds of aircraft and in the supercharged height for maximum performance, both of which demand high pitch angles from the airscrew, has resulted in very inefficient operation under sea-level conditions, the static revolutions per minute and the power output being held down considerably because of the coarse angle of attack of the airscrew blades. The present 10-deg. range of the two-pitch airscrew is found to be insufficient to meet the required conditions; it is considered that, to reap the full benefit from a variable-pitch airscrew, a variation in pitch angle of at least 20 deg. should be provided.

The view has been expressed, however, that in cases of certain lighter types of aircraft in which weight is of primary importance, the improvements in octane ratings of fuels have permitted engine manufacturers to obtain such increases in take-off power as to make the use of a variable-pitch airscrew unjustifiable on account of its extra weight and cost. The effect of the additional weight in the nose on maneuverability is considered to be a very important factor in aircraft of this category.

It is felt that the keenest attention should be paid to this weight problem, and that a lighter blade design would assist considerably by enabling a much lighter hub to be used. There is no doubt, however, that this type of airscrew has come to stay, and the ideal would seem to be the constant-speed infinitely variable pitch since it would enable the best use to be made of the power available under all conditions. The fact, however, that such an airscrew would allow the continuous use of maximum engine speed and power for each condition of flight, means a corresponding reduction in engine life and in periods between overhauls.

To determine the relative merits of ground-boosting with a fixed-pitch airscrew as opposed to increasing the engine speed at constant boost pressure with a variable-pitch screw, a series of full-scale, static-thrust tests have been carried out, the results of which are shown in Fig. 28.

These results demonstrate that the thrust at constant boost pressure increases with increasing engine speed, but that higher thrusts per unit horsepower are obtained at low boost pressures, the thrust/horsepower ratio decreasing under both constant speed and constant pitch conditions as the boost is increased. Considering the results another way and plotting ratio of thrust against ratio of power relative to some arbitrary



Figs. 28-30—Full-Scale Static Thrust Tests at Hucknall on Fury K.3586

Fig. 28—Engine Speed Vs. Static Thrust

Hamilton variable-pitch 11 ft., 0 in. diameter airscrew, 3-blader.

Fig. 29—Ratio of Power Vs. Ratio of Thrusts at Constant Speed with Varying Pitch Angle Compared with Fixed Pitch Case

Hamilton 3-blade variable-pitch air screw, 11 ft. 0 in. diameter. Ratios plotted relative to Hamilton airscrew, 32.5-deg. pitch at 0.7 of radius and zero boost pressure.

Fig. 30—Engine Speed vs. Thrust Per Horsepower

Hamilton variable-pitch 11 ft., 0 in. diameter airscrew, 3-blader.

base, as shown in Figs. 29 and 30, it is seen that high revolutions per minute and low boost pressures produce far greater increases in thrust than ground-boosting with a fixed-pitch screw. Thus for a Hamilton screw it was found possible to obtain an increase in thrust of 120 per cent for a power increase of 45 per cent at zero boost pressure by allowing the speed to increase from 1725 to 3000 r.p.m., whilst ground-boosting from the same conditions and keeping the pitch angle constant produced only a 55 per cent thrust increase for a power increase of 100 per cent.

Although not strictly relevant to the title of the paper, this section is included as an acknowledgment and confirmation of the theoretical analysis made by F. W. Caldwell[4] and presented to the S.A.E. in June, 1934.

Investigation of Powerplant Problems

The complexity of modern powerplant installations, the necessity for reducing to a minimum the losses consequent with dissipating waste heat from oil and coolant, and the efficient disposal of exhaust gases make impossible a separation between the engine and its installation. In realizing this fact it is also not sufficient only to advise the aircraft contractor as to the layout of his powerplant because there is necessarily a considerable delay between the conception of new designs and the results when manufacture, installation, and testing are each controlled by a separate organization.

We have, therefore, set up our own aerodrome and experimental flight section whose function is to explore all the problems associated with the development of a satisfactory powerplant with the ultimate purpose of supplying to the aircraft contractor the engine and all its adjuncts forward of the engine bulkhead. It also enables us to get new-type engines into the air quickly in order to get over rapidly the flight-testing troubles that are inseparable from a new-type engine before it passes out of our hands into those of the aircraft contractor. Although this section of the experimental department has been in operation but a comparatively short period, it is doing invaluable work on the new flight problems associated with the introduction of variable-pitch airscrews, high-temperature cooling, initial proving tests of new type engines, and general installation problems.

Photographs of three typical modern aircraft fitted with Rolls-Royce engines are shown in Figs. 31, 32, and 33.

Incidental Problems

It is suggested that other problems will arise when high-octane fuels come into general use. We have found even at the boost pressures possible on current D.T.D. 230 fuel that magnetos and spark-plugs are not up to requirements. Thus, on the fully supercharged Kestrel engine the full throttle height is 14,500 ft. and the corresponding boost pressure is 3.25 lb. per sq. in. The voltage requirement at the plug gaps is, therefore, being maintained up to altitudes where the flash-over conditions are very poor. By very careful attention to plug gaps and by using special low-voltage types we have been able to maintain satisfactory ignition up to 30,000 ft. An entirely redesigned ignition system will be necessary to deal satisfactorily with these conditions, which are becoming increasingly difficult as boost pressures and full-throttle heights increase.

Similarly blower efficiencies require stepping up, and it may be necessary to evolve new types. We are running single-stage centrifugal compressors at compression ratios up to 2.4:1 but it is very doubtful if this ratio can be extended very much more. In addition to the large temperature rise associated with such high ratios the power absorbed by the compressor becomes a prohibitive proportion of the engine power. It is, therefore, increasingly necessary to reconsider the exhaust-driven type of supercharger in relation to high-altitude ratings.

The recourse to higher outputs requires additional cylinders as indicated earlier in the paper, but the extra power must be transmitted to the airscrew shaft by the reduction gear which is difficult to divide up into a number of lightly

[4] See S.A.E. TRANSACTIONS, August, 1934, pp. 297-310: "Aircraft-Propeller Development and Testing Summarized", by Frank W. Caldwell.

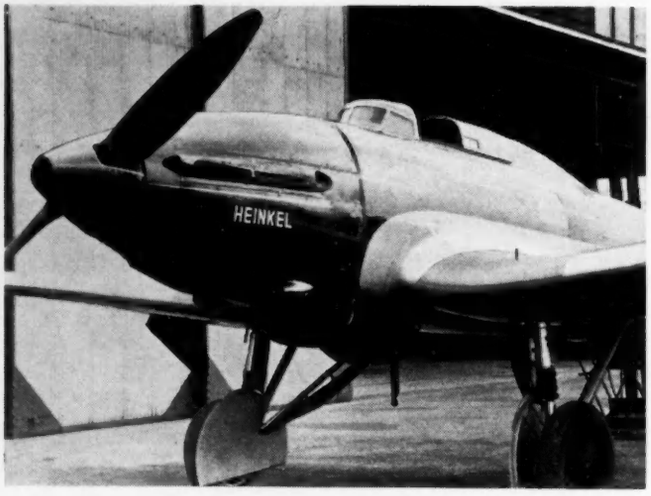


Fig. 31—Heinkel-Kestrel Engine

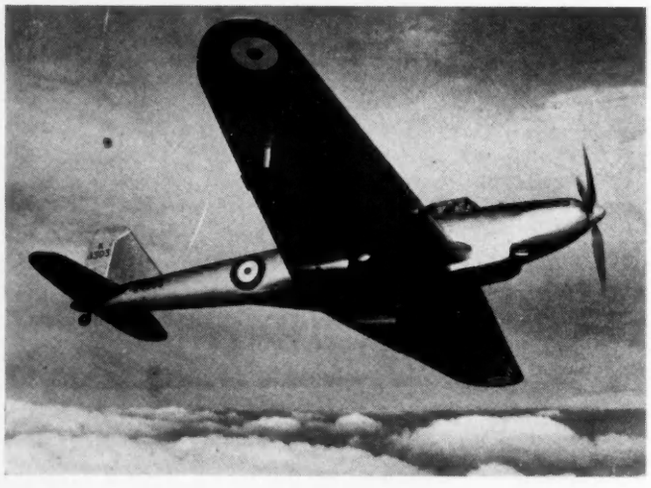


Fig. 32—Fairey "Battle", Merlin Engine

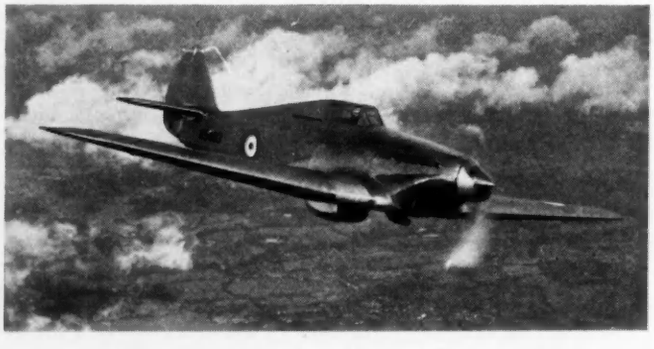


Fig. 33—Hawker Monoplane, Merlin Engine

loaded components. The lubrication of the heavily loaded surfaces may require separating them from the rest of the engine and use of an extreme-pressure type of lubricant. Present engine lubricating oils are holding down the temperature at which the engine can be operated and give rise to cooling difficulties. A step-up in oil temperature will allow a general step-up in engine performance, and we feel confident that the oil companies will ultimately meet this requirement.

We think that engine development during the period of incubation necessary to realize the advantages of the new fuels will result in the examination and evolution of new engine types operating at about 8:1 compression rate at 3000 ft. per min. piston speed developing brake mean effective pressures of the order of 220 lb. per sq. in. with a full-throttle height of 20,000 ft.

Our conclusions impel us toward the further development of liquid cooling for a solution of the engine type predicted previously, on the basis that all the features tending to restrict engine and aircraft performance can be overcome more satisfactorily by indirect cooling than by direct cooling.

Finally, I wish to record my appreciation of the assistance given by my colleagues in the Experimental Department in the compilation of this paper, particularly Mr. Ellor and his staff for their contributions on the aircraft problems and to the British Air Ministry for its permission to include the results of experiments made in its behalf and for photographs of machines and special installations.

Discussion

Differing Air-Cooling Factors Presented

—Kenneth Campbell
Wright Aeronautical Corp.

IN connection with the author's mathematical demonstration that cooling-drag power loss for an air-cooled cylinder increases prohibitively with increased power output, I should like to introduce some considerations which appear greatly to influence the results.

The author is quite correct in utilizing the well-substantiated relation that, for constant power output or constant heat flow, the temperature difference between metal at the base of the fins and the air varies as a fractional power of the mass flow of cooling air. However, it may also readily be demonstrated experimentally that, for constant mass flow, the temperature difference referred to previously varies as a fractional power of the horsepower and not at a rate even approaching the direct relation which the author assumes. This condition may be due to a leveling out of the declining temperature gradient of the fins with increase in heat loading, causing better utilization of the outer portion of the fins as cooling conditions become more severe. We find experimentally that the temperature difference varies as the 0.3 to 0.5 power of the brake horsepower with constant mass flow of cooling air. Like the exponent expressing the relation of mass flow to heat transfer, the value of this exponent apparently varies with fin design, with the heat loading, and with the velocity of the cooling air stream scrubbing the fin surfaces.

Viewed in this light then, the increase in mass flow required with increase in horsepower to maintain the same cylinder temperature is of a different order than the author's conception (a fractional power of the horsepower instead of approximately the square of the horsepower).

The paper also points out the possibility of increasing the cooling surface of liquid radiators as output of a given size of engine is increased so as to limit the increase in power necessary to force the cooling air through the passages, to a value proportional to the increase in engine power. For the comparable air-cooled engine, the paper refers to the cooling surfaces as "fixed" as outputs are increased which reference, of course, leads to the assumption that heat transfer can be increased only by increasing the velocity of the cooling air through the cooling passages about the cylinder. Quite to the contrary, the cooling surfaces of air-cooled cylinders in recent years have not been fixed in this sense, and the practice, which some aircraft engine manufacturers have applied for increasing heat transfer with a higher specific power output, is exactly that which the author suggests with regard to liquid radiators.

There are two basic methods for improving air-cooling of cylinders: (a) leaving the surface unchanged and increasing the air-stream velocity; (b) leaving the velocity unchanged (or decreasing it) and increasing the amount of surface. Until the cooling surface of the fins on modern air-cooled cylinders is increased beyond the point of diminishing returns, the increase in cooling-drag power loss resulting from the second method of improvement is only about one-third that obtained by the first method, for the same increase in heat transfer.

In the case of the Cyclone engine, the most recent increase in surface was 90 per cent, yet there is no evidence that the point of diminishing returns in fin development has been reached, nor has the possibility of further improvement in heat transfer by any means been exhausted. As

a result, the cylinders cool in flight at temperatures below 400 deg. fahr. at greatly increased power output and at the reduced air velocities obtained in the best rate of climb (about 3 in. of water static pressure drop from front to rear of the engine). This performance reflects a net cooling-drag power loss of just less than 1 per cent of brake horsepower, although the brake mean effective pressure for this condition is approximately 175 lb. per sq. in.

These considerations reflect an air-cooling-drag power loss in flight of only about 20 per cent of the values estimated in the paper.

The comment also is made that the use of the forward speed of the airplane for air-cooling at full power and at very high altitude will soon become impracticable except for racing types of aircraft. In so far as the author refers to stratosphere altitudes where the air temperature no longer declines as density declines, I am in agreement with his prophecy. However, up to the stratosphere level (about 35,000 ft.), the adverse effect on heat transfer of decline in mass flow of cooling air with increasing altitude is practically offset by the increase of temperature difference between cylinder and air due to the decline of the prevailing air temperature. Provided moderate cylinder temperatures are obtainable at sea level (400 deg. fahr. or less), the increase of these temperatures with altitude while maintaining best rate of climb should be slight if any. Published research results[1] offer some confirmation of this statement, as well as do unpublished information from flight tests to 25,000 ft.

Combining these findings with the reduced climbing speeds at which good cooling is now obtainable at rated power, I am not inclined to be discouraged by the outlook for dynamic pressure cooling at altitudes below 35,000 ft. Some take-off cooling problems, on the other hand, particularly in the case of slowly accelerating flying boats, do present a strong argument for induced air cooling.

Increased Number of Cylinders Favored

—W. A. Parkins

Pratt & Whitney Aircraft Co.

I AM particularly interested in what Mr. Wood has to say about size and number of cylinders and, in the main, agree with his conclusions. We feel that the power of engines of the type manufactured generally in this country has already reached such values that, from the standpoint of smoothness of operation, adequate cylinder cooling, and compactness, it is highly advantageous to employ more cylinders.

As Mr. Wood points out, more cylinders will increase the initial cost of engines, but the savings resulting from the less severe and more frequent power impulses on the wear and tear of the airplane and the gain in passenger-carrying revenue resulting from a more comfortable ride will, we believe, more than offset this higher first cost.

The opinion also has been advanced that more cylinders will not increase overhaul costs for the reason that the lighter explosions and more even torque will reduce parts replacements enough to make up for the labor of handling the additional cylinders, pistons, and rods.

However, regardless of whether the change works to increase or decrease maintenance expense, this factor is of minor importance for the reason that overhaul costs in this country have been brought down to such very low values. The average cost of parts replacements at overhaul on one of the well-known airlines in this country is very close to $78. The average number of man-hours per overhaul is 103. Thus, the total average overhaul cost is close to $150.

These engines operate consistently at slightly over 75 per cent of rated power and at a cruising brake mean effective pressure of 125 lb. per sq. in. The specific fuel consumption averages close to 0.50 lb. per b.hp-hr., and the engines are going 500 hr. between overhauls. I should like very much to compare these figures with similar figures on the best of the liquid-cooled engines operating in England.

Mr. Wood places considerably more emphasis upon frontal area than upon engine length. Furthermore, his comparisons of frontal area omit the oil-cooler and radiator protuberances of the liquid-cooled engine which, from observation, I should say adds to the frontal area a percentage that runs into two figures.

It appears to me that, in considering frontal area, one also should consider length. To go to extremes, it is almost as difficult to conceive of an engine 1 ft. in diameter and 20 ft. long being used in a single-engine aircraft as an engine 20 ft. in diameter and 1 ft. long. It appears to me that the age-old engineering rule of a good compromise applies here. What the good compromise is I am not prepared to say, but it probably depends a great deal upon the type and size of the airplane in which the engine is to be used.

Little, I believe, is known in this country about the relative drag of the radial air-cooled and the in-line liquid-cooled engine. I think, however, that it is the general opinion that there is little difference. I know of one comparative flight test which showed the air-cooled installation to be superior.

The idea that the air-cooled engine is unable to cool properly when supercharged at high altitudes is beginning to appear unsound. There is now enough known about high-altitude performance to say with a fair degree of certainty that the air-cooled engine can be made to cool properly without mechanical aids at altitudes in the vicinity of 20,000 ft. and with an output of 0.46 b.hp. per cu. in. displacement.

The comparative estimates of installation weights based on engines of 1000-b.hp. rating appear to favor the liquid-cooled engine too much to pass without comment. Had I made these estimates, the overall weights would have come out something like 760 lb. for the air-cooled engine and 930 lb. for the liquid-cooled engine. I should like to ask Mr. Wood why he believes that the cowling and exhaust manifold for the air-cooled engine should weigh so much more than similar parts on a liquid-cooled engine, and whether or not there are any liquid-cooled engines in normal operation in England whose cooling system weighs only 0.201 lb. per b.hp.

Author's Experience with Aromatic Fuels Confirmed

—Lieut. Frank D. Klein

U. S. Army Air Corps

MR. WOOD'S data on fuel tests are particularly interesting, especially the curves showing the effect of engine variables on allowable output at best-power and maximum-economy fuel consumptions. The unusual cases mentioned, where detonation increases as the mixture is enriched, have been duplicated at this Division in a high-output, single-cylinder liquid-cooled engine.

There is no question but that relative values of fuels change with changes in engine conditions, and that different engines themselves will fail to correlate as regards fuel performance. This condition is particularly true in the case of highly aromatic fuels. In connection with the superior performance reported by Mr. Wood for aromatic fuels, it is interesting to note that in a high-output single-cylinder liquid-cooled engine at this Division, a highly aromatic fuel of 92-octane number by the Air Corps method of knock test allowed 32 per cent higher output than an iso-octane blend of 100-octane number, both at the same point of light detonation. This engine operated under conditions differing greatly from present service-type engines. Since aromatic fuels are known to give higher cylinder temperatures, in the case of air-cooled engines, than other types of fuel operating under the same degree of audible knock, it is practically certain that this 92-octane aromatic fuel would not show to such marked advantage in an air-cooled engine, using cylinder temperature rather than audible knock as the criterion of limiting output. The value of aromatics in both air-cooled and liquid-cooled engines with variable mixture temperature, and in full-scale engines, will be determined later.

Because different engines rate fuels differently, it is necessary to develop knock-test methods that will rate all fuels in the order of their average value in existing service-type engines. On this basis, present knock-test methods rate all fuels, including aromatics, closely in accordance with their full-scale performance, and lack of correlation between different engines has not in the past been serious. There is little doubt, however, but that knock-test methods will have to be revised to some extent, whenever radical changes in service-type engines take place.

Radiator Proportions Discussed

—W. Worth

Materiel Division, U. S. Army Corps

STUDIES at the Materiel Division indicate that, when external drag can be neglected, the same surface arranged with a large frontal area and shallow depth will dissipate more heat for a given available pressure.

If this condition is true, then it seems that practical considerations and available space should be the determining factors in all cases where the radiation is located within the structure; which conclusion is not consistent with the definite values plotted by Mr. Wood.

The weight figures of 10 and 25 lb. allowed for the oil-cooling required by liquid-cooled and air-cooled engines are much lower than those experienced in this country. It would be interesting to know what methods are used for reducing the external oil-cooling, and the benefits derived from crankcase fins.

[1] See S.A.E. TRANSACTIONS, November, 1935, pp. 401-410; "Evaluation of Variables Influencing Cooling of Engines", by Kenneth Campbell.

The Development of "Proferall" Cast Camshafts

By D. J. Vail

Campbell, Wyant & Cannon Foundry Co.

DEVELOPMENT of cast camshafts at the Campbell, Wyant & Cannon Foundry Co., starting in 1924, proceeded slowly until a material was developed that met all requirements from metallurgical, engineering, and manufacturing standpoints.

"Proferall," the name given this material, means processed-ferrous-alloyed iron made by the duplexed-electric-furnace process. Camshafts of this material have a Brinell hardness of 262-293, as cast.

A series of tests, equivalent to runs of 46,560 miles, showed that both chemical analysis and hardness affect camshaft-gear wear. Comparative wear tests on bearings showed more than three times as much wear on steel camshafts as cast ones. Other tests showed the cast shafts expanded less than those of steel.

After describing foundry processes the paper concludes by summing up the advantages of cast camshafts, such as the smaller cost of patterns as compared with forging dies and the elimination of heat-treating, copper plating, carburizing, and hardening.

CAMPBELL, Wyant & Cannon Foundry Co. started development work on cast camshafts in 1924. From that time until 1927 considerable experimental work was done, but the greater part of the experimentation was confined to molding and the method of making the casting in the foundry. Little time or thought were given to finding a suitable metal for camshafts.

For the next two years this work was slow due to two reasons:

First, increase in production on regular lines of castings left little time or space in the foundry for this development program. However, at no time did the development work on camshafts stop entirely. The second cause for delay was hesitancy on the part of the engineers to accept the new idea as a sound and practical proposition.

Had the engineers given the foundry more encouragement at this time, the cast camshaft would have been in production at least two years sooner than it was. It is this trait in engineers that gives us value in our automobiles of today. They take few chances and hesitate to approve any new material until they have tried it out thoroughly; they must be sure before releasing any material that goes into the production of an automobile.

In the latter part of 1929 when business slowed up on regular lines of castings, Campbell, Wyant & Cannon Foundry Co. started to push the development work on cast camshafts, and definitely decided to make a camshaft and develop a material that would meet all requirements from metallurgical, engineering, and manufacturing standpoints.

Having most of the problems of molding well in hand, more time and thought could be given to the developing of a suitable material for cast camshafts. Hundreds of camshafts were made and tried out, all having different chemical analyses and hardnesses. Some failed due to excessive wear on the gears and bearings; others did not have the required strength and hardness; and others had proper hardness, strength, and wearing qualities, but could not be sold due to excessive material costs.

After many discouraging months of testing, the following material proved to have all the necessary qualifications to make a good camshaft from the engineering and metallurgical standpoints, as well as a material that could be machined in keeping with the speeds and feeds required of any material used in this day and age of high production:

Analysis Specification

Total Carbon	3.10-3.40
Silicon	2.10-2.40
Sulphur	0.10 maximum
Phosphorous	0.20 maximum
Manganese	0.50-0.75
Chromium	0.75-1.00
Nickel	0.20-0.40
Molybdenum	0.40-0.60

The metal from which these camshafts are made is a

[This paper was presented at the Annual Production Meeting of the Society, Detroit, April 21, 1936.]

duplexed-electric-furnace alloyed iron, known to the trade as "Proferall" (processed-ferrous-alloyed iron). The metal, as charged in the cupola, consists of grade A scrap and selected pig iron. The only alloy added at this point is chromium in the form of chromium briquettes. The melted metal is tapped from the cupola at approximately 2800 deg. fahr., and then transferred to an electric furnace where nickel and molybdenum are added. Here the metal is refined, superheated, and checked for temperature and chemical analysis. Duplexing is done in a furnace bath of four to five tons, thus promoting uniformity of the final product.

A sample for hardness, taken from each ladle of iron tapped from the furnace, is tested and approved before the camshaft castings are poured. Several times throughout the day camshafts are taken from the production line and checked for hardness. In addition, arbitration bars as well as camshafts are taken at regular intervals through the day for transverse tests. This procedure gives us a check on the metal as well as a check on the camshaft itself.

The average camshaft, made to the foregoing chemical analysis by our method, will have a Brinell hardness of 262-293, as cast. We have found from extensive testing that 262 Brinell is the low limit that can be used successfully on a camshaft gear of this material. Softer than this, one is very apt to have considerable trouble with gear wear.

One of the first and major difficulties that had to be overcome was the possibility of gear wear. With this in mind, we developed a machine for testing camshaft gears which is designed so that we can run a set of gears and have any oil pressure desired from 1 to 250 lb. per sq. in. The oil pumps used on this testing machine will deliver 1 gal. of oil per min. with the camshaft running at 2000 r.p.m.

Wear on a camshaft is not always a direct result of hardness of material or of lack of hardness. There are many other elements that enter in and cause wear. Many tests were made to determine the amount of wear on gears of different compositions. From a study of the five tests that follow, it will be seen that both chemical analysis and hardness must be considered—one without the other will not give a material that will withstand gear wear.

Composition

The chemical analyses were taken from each sample of camshaft gear.

	Gear Number				
Constituent	1*	2*	3	4	5*
Total Carbon	3.22	3.16	2.98	3.05	3.22
Graphitic Carbon	2.42	2.59	2.23	2.40	2.42
Combined Carbon	0.80	0.57	0.75	0.65	0.80
Silicon	2.02	2.68	2.72	2.54	2.02
Sulphur	0.140	0.080	0.145	0.147	0.140
Phosphorous	0.16	0.17	0.17	0.14	0.16
Manganese	0.74	0.61	0.60	0.60	0.74
Chromium	0.77	0.67	0.92	0.21	0.77
Nickel	0.39	0.78	1.53	...	0.39
Molybdenum	0.45	0.55	...	...	0.45
Copper	...	...	...	2.63	...
Brinell Hardness	286	255	272	294	286

*Composition being used in our camshafts and gears.

Results of Test

Each gear was run 46,560 miles with the oil pressure at 60 lb. per sq. in. The following are the results of the tests:

Gear Number	Thickness of Tooth at Pitch Line, in. Before	After	Amount of Wear, in.
1	0.130	0.128	0.002
2	0.130	0.125	0.005
3	0.130	0.116	0.014
4	0.130	0.1225	0.0075
5	0.130	0.119	0.011

Gear No. 3, as shown by the above tests results, was subject to the most wear, while gear No. 1 was the most satisfactory.

The mating gears, or the oil-pump and distributor gears, used in these tests were made of the same material as the gears that they were run with, except one used with sample No. 5, which was a hardened-steel gear. The wear on oil-pump gears Nos. 1-4 was the same as that on the camshaft gear. Sample No. 5 showed 0.005-in. wear on the oil-pump gear. It can be seen from the test of sample No. 5 what happens to the camshaft when a hardened-steel gear is used to drive the oil pump and distributor.

Oil-pump gears made of steel, carburized and hardened, will not work well with the cast camshaft unless the loads are extremely light. This difficulty is due to the roughness of the steel gear as cut. When the gear is hardened, the lines left by the gear hob stand out and act like a file on the soft cam gear, thus causing excessive cutting or wear. This difficulty was overcome by using combination No. 1, which is two gears of the same material.

Exhaustive wear tests on bearings were made, both in the laboratory and in actual motor-service tests on the road, ranging from 50,000 to 125,000 miles at high speeds. Calibration showed maximum wear to be less than 0.0003 in. in

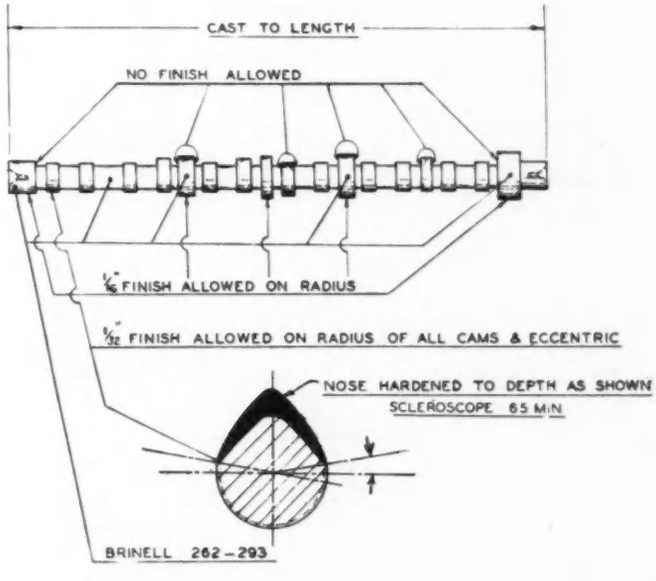

CAST TO LENGTH
NO FINISH ALLOWED
1/16" FINISH ALLOWED ON RADIUS
1/32" FINISH ALLOWED ON RADIUS OF ALL CAMS & ECCENTRIC
NOSE HARDENED TO DEPTH AS SHOWN
SCLEROSCOPE 65 MIN
BRINELL 262-293

Fig. 1—Proferall Cast Camshaft

all cases on the cast shaft. Steel shafts in equal mileage tests and under the same conditions showed 0.001 to 0.0015 in. of wear.

One cause to which wear can be attributed is expansion of the shaft material under service conditions. Expansion of Proferall is less than that of steel and, for this reason, the allowances provided in bearing fits for lubrication are not taken up by expansion. Therefore, the film of lubricating oil serves its purpose continuously in protecting both the shaft and the bearings in the motor. The following figures compare expansion of Proferall and expansion of S.A.E. 1020 steel. Two bars were used, 1 in. in diameter by 8 in. long. The hardness of the Proferall bar was 293 Brinell, of the steel bar, 235 Brinell. They were heated to 100 deg. fahr., then calibrated and allowed to cool to room temperature before heating to the next step.

Proferall Brinell Hardness – 293			1020 Steel Brinell Hardness – 235		
Temperature, deg. fahr.	Diameter, in.	Expansion, in.	Temperature, deg. fahr.	Diameter, in.	Expansion, in.
68	1.000		68	1.000	
100	1.000		100	1.0003	0.0003
150	1.0002	0.0002	150	1.0005	0.0005
200	1.0004	0.0004	200	1.0006	0.0006
250	1.0005	0.0005	250	1.0011	0.0011
300	1.0010	0.0010	300	1.0015	0.0015
350	1.0012	0.0012	350	1.0016	0.0016
400	1.0015	0.0015	400	1.002	0.002
450	1.0020	0.002	450	1.0023	0.0023
500	1.0023	0.0023	500	1.0027	0.0027
Cooled in air to 68 deg. fahr.					
68	1.000		68	1.000	

Some of the outstanding features of Proferall will be seen by the tensile and torsional tests that were made at the Michigan State College.

Torsion Test of Alloy Iron for
Campbell, Wyant & Cannon, Muskegon, Mich.

Test number	5
Bar number	119 (Proferall 3X)
Bar diameter, in.	1.0
Test length, in.	6.0
Polar moment of inertia $= J = 1/32\,\pi\,D^4$	0.09816
Factor $= C/J$ ($C =$ radius $= \frac{1}{2}$ in.)	5.093
Torsional moment at proportional limit, in. lb. (Taken from curve)	6,750
Torsional moment at rupture, in. lb.	16,100
Stress at proportional limit, lb. per sq. in. $S_1 = M_1\,(C/J)$	34,150
Modulus of rupture $S_2 = M_2\,(C/J)$	82,000
Troptometer reading at proportional limit, 1/50 in. (From curve, 1 in. = 5 deg. twist in 6 in.)	35
Deformation at surface, in. per in.	0.0051
Modulus of rigidity (E = stress/strain)	8,800,000

Tested Jan. 13, 1934.
By W. E. Reuling.

Tension Test of Alloy Iron for
Campbell, Wyant & Cannon, Muskegon, Mich.

Test number	10
Bar number	119 (Proferall 3X)
Bar diameter, in.	1.0
Test length, in.	6.0
Test area, sq. in.	0.7854
Load at proportional limit, lb. (From curve)	29,500
Maximum load, lb.	42,350
Fiber stress, lb. per sq. in.	
At proportional limit	37,500
Maximum	53,800
Extensometer reading at proportional limit, 0.001 in. (From corrected curve)	32
Stretch in 6 in. at proportional limit, 0.001 in.	16
Unit strain, in. per in.	0.00266
Modulus of elasticity (Unit stress/unit strain)	16,100,000

Tested Jan. 13, 1934.
By W. E. Reuling.

Cast Proferall camshafts are molded very much like any other casting of the same size and dimensions. Patterns are mounted on heavy-duty molding machines, and molds are made at the rate of approximately 45 per hr. This rate, of course, depends upon the size of camshaft and the number of castings in each mold. Molds made on these machines are placed on a power conveyor where they are closed and poured continuously during the shift. Castings are left in molds for approximately 45 min. after pouring, before shaking them out, after which operation they are placed on a chain conveyor that takes them to the cleaning room.

The first operation in the cleaning room is to remove the gates and risers. Castings are then placed on a roller conveyor and go to a specially built shot blast. This blast will clean approximately 600 camshafts per hr. The next operation is to remove the excess metal, if any, from each end of the shaft. After the camshafts are snagged or milled to length, fins and parting lines are removed from the body of each shaft. This operation is done with a small flexible grinder using a wheel 3/16 in. thick by 3½ in. in diameter.

Camshafts are now cool enough so that they can be handled with the bare hands. It is quite necessary that the temperature of the castings be less than 150 deg. fahr. before the next two operations are performed, the first of which is to test for impact. Each camshaft is placed in a fixture and a 35-lb. weight dropped 14 in., the weight striking the casting in two places. After the impact test the casting is checked for hardness of gear blanks and bearings. This operation is done by means of a Gogan hardness-testing machine. Camshafts are then file-tested for hardness of cams and centered. Centers are held to ± .005 in. for depth in relation to some center cam. Castings next go to the straightening fixture, where they are straightened if necessary. Camshafts also are inspected on centers at this time and must run within 1/64 in. for straightness in order to pass inspection.

One of the questions most frequently asked is: "Can cast camshafts be straightened?" Yes, they can, and much more rapidly than steel shafts. Shafts should be sprung by means

of a bar and peened slightly at the opposite side from the bar to hold in place. Peening is necessary due to the fact that the elastic limit is close to the ultimate strength of this material.

From the standpoint of manufacturing expense in equipment, tools, and production costs there are many advantages to be credited to the use of Proferall as a camshaft material. Primary in the savings which accrue is the cost of forging dies as compared with pattern expense. Patterns can be completed in less time and at much less cost than dies for forgings. There also is an added advantage in that either minor or major changes in the shaft design can be effected without delay and at slight cost.

The use of Proferall eliminates many of the operations necessary on drop-forge shafts. The omission of heat-treating, copper plating, carburizing, and hardening – these features eliminate the maintenance of heat-treating furnaces, plating equipment, and the floor space that they occupy. The additional finish required on a rough forging is unnecessary. Therefore, the milling lathe operation for rough turning the contours of cams and eccentric is dispensed with, also the machines necessary for spacing cams and bearings and turning the body of the shaft to size.

Proferall camshafts are cast with 1/16-in. finish on a radius of bearings for rough-turn and finish-grind. Cams and eccentric faces have 1/32 in. on a radius for rough- and finish-grind only. See Fig. 1. There is no finish allowed on the sides of bearings, sides of cams, or body of shaft and, in most designs, there is no finish left on the ends of the shaft. Proferall camshafts resist the wear equally well of tappets made of any material, steel tappets causing no greater wear on cams than those made of chilled iron. There is also an advantage in that tappet noises are much less with the cast camshaft. This is a feature very much appreciated by the engineer as well as the car owner.

Discussion of Vail Paper

Machining Hudson Cast Camshafts

—Hudson Motor Car Co.

CAMSHAFTS made of cast alloy have been in production at Hudson Motor Car Co. since the introduction of the Terraplane Six in the summer of 1932. They have been as satisfactory in service as the forged shaft formerly used. However, there were difficulties to overcome in the first-production castings, and several changes made since that time have definitely improved the machinability of the castings. The first part of this paper will need to mention certain features of the casting itself as they affected the shop.

In the early form the cams were chilled by inserting chills in a mold by hand. This method was hard to control because, if chills were not properly placed, cams would have excessive stock on one side which sometimes caused soft cams after grinding due to the grinding away of most of the chilled surface on one side. This original method of chilling also caused a great deal of trouble due to center-bearing and gear-diameter sidewalls being too hard to machine satisfactorily without added tooling expense.

An erratic spacing of the cams and a lack of uniform stock removal on the center-bearing and gear sidewalls were eliminated by dimensioning a central cam with the rear center and having the foundry maintain it within limits of ±0.015 in. This design removed the need for facing the rear end of the shaft and re-centering, which also eliminated the sidewall machining of the center-bearing.

A changed method of hardening the cams was the foundry's most important contribution to economy in machining this new shaft. This improvement eliminated the objectionable shift in the cams and confined the hardening to these points. It was then possible to reduce the amount of stock left for machining and, because of the uniform thickness of metal removed, the rejections for soft cams and blow holes were reduced to 1 per cent at the present time.

Before relating the production operations as performed on the cast shaft, it may be well briefly to describe its construction.

The oil-pump and distributor-drive gears and a fuel-pump cam are cast integral with the shaft. The Hudson "8" shaft has five main bearings, while the Terraplane "6" has three. The timing gear is mounted on the front-bearing flange, and the rear of this flange serves as a thrust face.

Cams and the entire fuel-pump eccentric are hardened to a minimum depth of 1/16 in. They are held to a scleroscope reading of 75 and are checked 100 per cent for hardness when the shaft is finish-inspected. The heel of the cams and all other unchilled surfaces scleroscope about 55, or 277 Brinell.

1/16 in. of stock is allowed for machining on the line bearing and gear diameters, and from 1/32 to 3/64 in., on the chilled cams. The location of the cams is maintained with the flange thrust face within a limit ±1/32 in. The cast width of the cam itself is 3/8 in. plus 0.00 in., minus 1/64 in., and it has a 3/32-in. radius where the sidewalls meet the shaft diameter.

In changing from a forged to a cast shaft, the following operations were eliminated:

(1) Centering.
(2) Rough straightening.
(3) Machining the cam and line-bearing sidewalls for correct spacing.
(4) Rough form-turning the cams.
(5) Facing rear end of shaft to length and re-centering.
(6) Copper plating.
(7) Carbonizing and hardening.
(8) Straightening after hardening.
(9) Reduction in amount of finish-straightening.

This paper does not purport to set up the present Hudson equipment or sequence of operations as being the most efficient for machining cast-alloy shafts. Rather it will be seen that the reduced number of machining operations is performed on the same equipment as was formerly used, and that only a rearrangement is necessary so that the flow of work continues to move in a straight line through adjacent machines.

The shaft is received from the foundry centered and straightened.

Hudson camshaft machining operations begin by snagging any irregularities or high spots that would be an interference point on the body of the shaft or on the cam sidewalls. This is more of an inspection operation as the shafts, for the most part, are free from any interference points.

The second operation is to grind two steady-rest diameters (on a 10- x 36-in. Norton grinder, production, 34½ camshafts per hr.).

The third and fourth operations are to rough-turn Nos. 1, 3, and 5 line bearings, (on a 3½ - x 60-in. Low-Swing lathe) and Nos. 2 and 4 line bearings and the flange outside diameter on another. ("J" stellite tools are used, and the production for one operator on two machines is 29½ pieces per hr. per machine – 0.021-in. feed at 92 r.p.m.)

The fifth operation is to straddle-face the timing-gear flange, two gear diameters, and chamfer all line bearings (on a 4 x 60 Low-Swing, using stellite tools. The production for one operator, on two machines, is 25½ pieces per hr. – 0.0077-in. feed at 62½ r.p.m.).

The sixth operation is to drill, ream, countersink, and counterbore the 0.500 in. diameter thrust-plunger hole in the timing-gear flange and finish-face the flange (on a No. 6 long-base Warner & Swasey Screw Machine, using a 0.010-in. feed at 480 r.p.m. The production is 34½ parts per hr., and Carboloy tools are used for the facing operation).

The seventh operation is to finish-turn and chamfer the timing-gear flange outside diameter and undercut the front main bearing at the flange thrust face, for grinding wheel and thrust-washer clearance. (This operation is done on a 3½ - x 60-in. Low-Swing lathe using Carboloy tools. The production is 48½ per hr. using 0.0128-in. feed at 83 r.p.m. for turning).

The eighth operation is chamfering the rear end (on a No. 4 long-base Warner & Swasey screw machine at 194 pieces per hr.).

The ninth operation is to mill a ¼-in. driving slot in the timing-

gear flange front face to serve as a driver when grinding. (This operation is done on a No. 2 Kent Owens hand mill at a production of 272 pieces per hr.).

The tenth and eleventh operations are drilling the three timing-gear bolt holes in the flange face (on a 24-in. Aurora drill press with a 0.0039-in. feed at 640 r.p.m.) and countersinking them. The production is 74½ parts per hr.

The twelfth operation is to finish-turn the five line bearings and two gear diameters, removing 0.035 in. of stock, (with a 0.0162-in. feed at 108 r.p.m. The production is 33 per hr.). A cooling lubricant is used on some of the rough operations, but it is not necessary as Carboloy and J stellite tools are used and the revolutions per minute are low.

The thirteenth operation is tapping the three bolt holes (on a No. 10 Gaterman tapper at 113 pieces per hr. A three-flute ground-thread H.S.S. tap is used, and the r.p.m. is 420).

The fourteenth operation is polishing the centers before grinding operations.

The shafts are checked for main-bearing alignment before rough-grinding the bearings, and they are straightened when necessary. Only the longer Hudson "8" camshaft requires this operation.

The line bearings are next rough-ground on 10- x 36-in. Norton grinders in the following order:

Center and rear bearing (Production is 43½ per hr. – 50N Alundum wheel used).

Front and rear intermediate (Production is 49 per hr. – 50N Alundum wheel used).

Front bearing outside-diameter and flange thrust face (Production 68 per hr. – 1936 C.M. wheel) 0.025 to 0.030 in. of stock is removed in the rough-grinding.

The final 0.005 in. of stock is removed in finish-grinding on 10- x 36-in. Norton grinders as follows:

Center and rear bearing (Production 45 per hr. – 50N wheel used).

Front and rear intermediate bearings (54½ per hr. – 50N wheel used).

Front bearing outside diameter and flange thrust face (42½ per hr. – 50N wheel used).

The front face of the timing-gear flange is then ground to square it up with the shaft centerline. (The production is 122½ per hr. – A 36K Crystalon wheel is used for this operation).

The shaft is now ready for rough- and finish-grinding the cams on 5- x 40-in. Landis automatic grinders. A 60 P6T Alundum wheel is used for roughing, and a 50 M5B Alundum wheel, for finishing. One operator runs two machines, roughs on one machine and finishes on the other. The production of finished shafts is 6 per hr.

A cooling lubricant is used for all the grinding operations. The cams clean up after grinding away only 0.015 in. on a side, so the maximum case is preserved.

The twenty-fifth operation is to cut the double-thread oil-pump-drive worm gear (on a Hanson & Whitney thread miller – 0.040-in. feed at 140 r.p.m.).

The sharp ends of the worm are next milled off (on a No. 1 Kent Owens hand milling machine). The production on both operations is 23½ per hr. for one operator.

The twenty-seventh operation is cutting a helical distributor-drive gear (on a No. 12 Barber Coleman, using a vertical feed attachment and special work slide for the double over-arm. Production is 30 per hr. (15 per machine) with an 0.0019-in. feed at 80 r.p.m. – The depth of the cut is 0.154 in.).

The worm and helical gears are then checked for concentricity and back-lash, with master gears in a special fixture, after which all burrs are removed and the main bearings and cams are polished on a speed lathe. (The production for this operation is 34½ per hr.)

The last operation is finish-straightening, when necessary. The main bearings are seldom out of alignment over 0.003 indicator reading, and they are straightened to 0.001 indicator reading.

All shafts are now inspected on the bench. The ground diameters are checked with micrometers for size, and then the concentricity of the cams and bearings, as well as the squareness of the timing gear flange face, is indicated.

Main-bearing diameters are held to a 0.001 in. limit.

The concentricity and alignment of these bearings is held to a 0.001 indicator reading. The squareness of the flange face is held to the same limit. The backs of the cams are held concentric within 0.0015 indicator reading. The limit for out-of-round and taper on the bearings is 0.0005.

The angle of all cams is maintained with the centerline of No. 1 exhaust cam within 0.002 in. and the timing-gear bolt-hole location is also held within ±0.005 in. with this cam. The contour and the angular relation of the cams are checked on a Pratt and Whitney Comporator. This checking fixture employs a master camshaft and follower roller arrangement by which production shafts can be compared with the master shaft quickly and the error shown in thousandths of an inch on the indicator mounted on the follower arm. No deviation from the master shaft in excess of 0.0015 in. is permitted.

Case Presented for Modern Forged-Steel Camshaft

—R. E. W. Harrison
Chambersburg Engineering Co.

AS a description of a long and successful struggle against extremely difficult conditions and engineering specifications, the paper is very informative, but there is a question, with which Mr. Vail does not deal, that confronts most of us who are dealing with the designing problem involved in this matter. This problem is the economic one measured in terms of cost and performance.

The extension of the use of cast components for such elements as crankshafts and camshafts probably is influenced by the extreme age of much of the forging equipment in use in the United States today. The American Machinist survey of 1935 showed that, whereas 65 per cent of machine tools at large are over ten years old, 85 per cent of the forging hammers in use are ten years old or even more ancient.

When forging equipment reaches this age its inaccuracies result in machining surpluses that increase costs; no doubt this factor has had material bearing in establishing the favorable situation attributed to the cast camshaft.

Another question refers to the desirability of an X-ray examination of every camshaft produced by this method. Is this form of inspection used, what does it cost, and what is the percentage of rejects?

Broken forged-steel camshafts have become practically unknown during the last few years, and it would be interesting to know if experience with the cast job shows that the new camshafts are standing up with the same freedom from breakage.

With the exception of the turning of the sides of the gear flange, there is no reason why forged-steel camshafts should not go direct to the grinding machine rather than to a cam-contour shaping, milling, or turning machine.

Inspection of camshafts in the state in which they now go from the forge shop to the machine shop indicates that the unduly large surplus of metal on these products has probably done more than anything else to enhance the favor accorded the cast job. The writer feels that the balance sheet of costs applicable to both methods or types of production would have been a different picture had the utmost advantage been taken of the possibilities of modern forging equipment before the switch-over was made to the cast product.

This situation does not in any way diminish the credit which is due the very fine metallurgical and production achievement with which Mr. Vail's paper deals.

Laminated Safety Glass

THE principle of laminated glass, as such, is old, dating back to the latter part of the nineteenth century; but, like many other industries, during its early stages little money or well-directed scientific and engineering effort was expended by those closely associated with it, with the result that four or five years ago the industry was still in its infancy and required corresponding treatment.

For the idea of laminated safety glass as we know it today, as far as public records are concerned, the honors go to an Englishman, Wood, who in 1905 obtained a British patent which describes a method for safety glass manufacture by the use of Canada balsam for cementing a sheet of transparent celluloid between two sheets or plates of glass. The Safety Motor Screen Co., Ltd., made samples of safety glass in this manner and exhibited them at the Spring Motor Show in England in 1906. Because of the high cost of materials, the general unsatisfactoriness of this product, and the small demand, Wood's venture was without success and the patent was allowed to lapse.

The first man to capitalize on the idea of laminated safety glass was a Frenchman, Benedictus, who obtained French and British patents in 1910. Benedictus named his product "Triplex" and employed the same general principle as Wood, except that he proposed gelatin instead of Canada balsam as the bonding adhesive for glass plates and celluloid. Benedictus introduced the manufacture of Triplex safety glass in 1912 in England where production started in 1913. The new industry received an enormous impetus during the World War when laminated glass was used for the manufacture of gas mask lenses and goggles, and for automobiles and airplanes.

Excerpt from the paper "Laminated Safety Glass" presented at the S.A.E. Production Meeting, April 24, 1936, by R. H. McCarroll, Ford Motor Co.

Extreme-Pressure Lubricants Testing

By George L. Neely
Research Engineer, Standard Oil Co. of Calif.

THIS paper describes briefly why extreme-pressure lubricants are necessary and what machines most commonly have been used to evaluate them.

Examples of differences in test results caused by differences in operating conditions, by the type of machine used, and by the method of installation of the machine are presented.

The new S.A.E. Extreme-Pressure-Lubricants Tester is described, and the probable advantages of the machine are discussed.

Other requirements of commercial extreme-pressure lubricants are listed.

In conclusion, the statement is made that the new machine is believed to constitute an approach to the gear problem that is more sound fundamentally than that of any of the other machines that have been used. In the older machines one or more of the test surfaces was always stationary while, in the S.A.E. Tester, both surfaces are in motion and a combination of sliding and rolling friction approaching that of gearing may be achieved.

In spite of this progress, however, the problem is by no means solved and a true solution can be achieved only by correlating the S.A.E. Tester with actual service tests.

IT is the purpose of this paper to review briefly extreme-pressure-lubricant testing, to present some interesting results obtained in our laboratories, and to describe the new S.A.E. Extreme-Pressure-Lubricants Tester. The development of new types of metals that are very much higher in tensile strength than the best steel available several years ago and the development of faster and more powerful engines have resulted in a great increase in the unit loading of the oil film on the gear teeth of modern cars. Also, the development of hypoid gears, which have certain advantages for structural reasons, has further increased the severity of the condition by adding a high rubbing speed of one gear tooth against another at high unit loading. Hypoid gears and several other types of gearing cannot be lubricated successfully with ordinary mineral oils, but require special lubricants which are known as extreme-pressure or E.P. lubricants. These lubricants generally contain certain compounding materials containing sulphur, chlorine, lead, or combinations of these and other chemicals.

With the development of gearing that requires special lubricants, several different types of laboratory testing machines were built in order to determine the film strengths of E.P. lubricants without resorting to full-scale gear-test equipment. These laboratory machines differed in design and method of operation; consequently, there was little or no correlation in the test results. As a result, a gear-oil manufacturer would select one of these machines and a set of operating conditions that he or others felt would correlate with gear-set performance, and would then proceed to develop a product of exceptional merit for lubricating the testing machine under the test condition employed. Unfortunately, the car manufacturers did not agree in regard to which testing machine and operating condition was preferable and, hence, the gear-oil manufacturers found it impossible to meet the extreme-pressure specifications of the various car manufacturers with a single type of E.P. product.

It should be said to the credit of the automotive industry that due warning was served to the petroleum industry of the

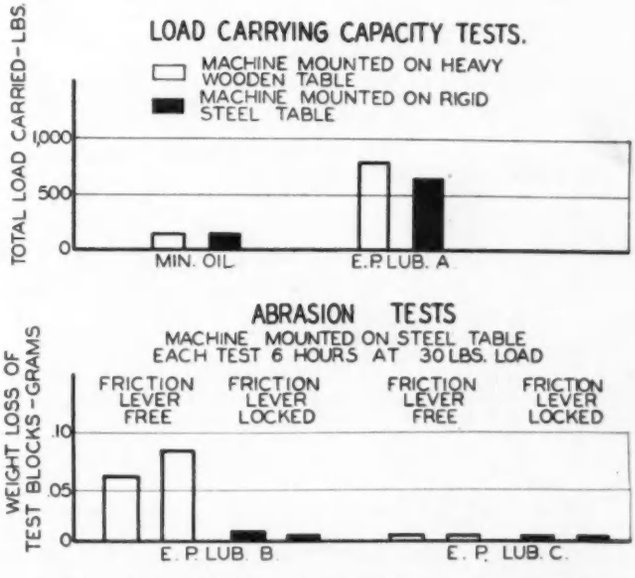


Fig. 1—Timken Machine—Effect of Rigidity of Test Machine on the Performance of Extreme-Pressure Gear Oils

[This paper was presented at the Pacific Regional Meeting, San Francisco, Nov. 19, 1935.]

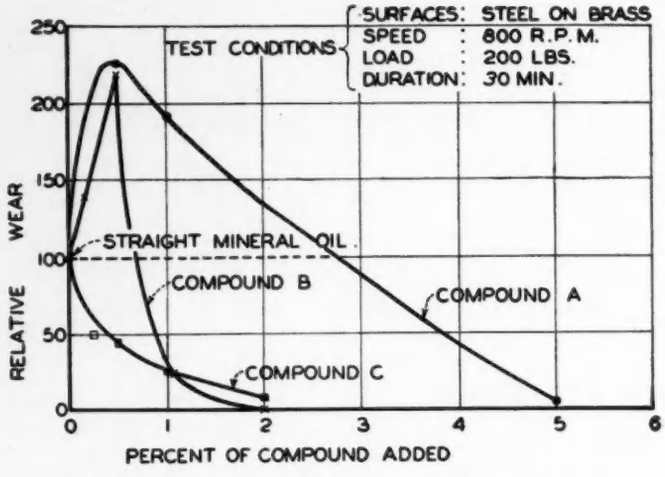


Fig. 2—Timken Machine—Effect of Concentration of Addition Agent on Wear

intended use of gearing requiring extreme-pressure lubricants. To meet this situation an S.A.E. subcommittee on extreme-pressure lubricants was formed for the purpose of investigating the subject and developing, if possible, a standard method for evaluating extreme-pressure lubricants using one of the machines then available. This testing method was, of course, to be substantiated by service data. This subcommittee secured the aid of the U. S. Bureau of Standards to test four well-known types of machines that had been built for the purpose. Eight different types of typical E.P. lubricants and a straight mineral oil were tested in each of the machines under a wide range of operating conditions. The machines used in this investigation consisted of the following: (1) Almen machine, (2) Timken machine, (3) Floyd machine, and a (4) large General Motors machine.

The preceding machines were operated at various conditions of speed and temperature and, from the results obtained, it was concluded that no one of the machines under any possible set of conditions would rate the lubricants in accordance with their true service values.

Some interesting results obtained in our laboratory with the Almen machine, the Timken machine, and a small worm-gear machine are presented to illustrate the complexity of the problem, the lack of correlation among results of these machines, and the importance of operating conditions.

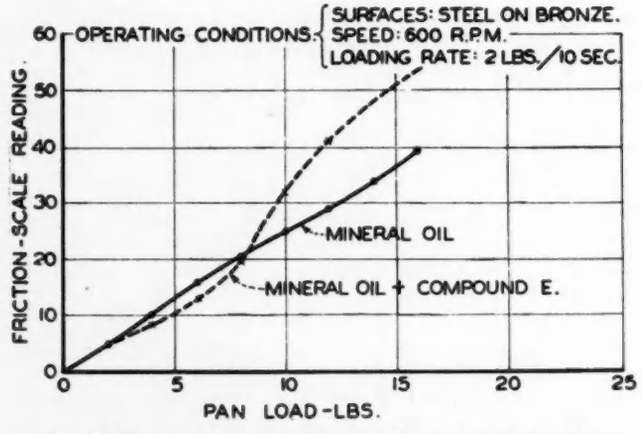


Fig. 3—Almen Machine—Effect of Intensity of Loading on Friction-Reducing Value of a Compound

Fig. 1 shows how vibration affects the Timken machine results. When this machine first was installed in the laboratory, a certain type of lubricant consistently showed a load-carrying capacity of about 800 lb. when the machine was mounted on a heavy wooden table. When the machine was changed to a rigid steel table, the load-carrying capacity fell off to about 650 lb. for the same lubricant. The rigid mounting reduced, in general, the load-carrying capacity of all types of E.P. lubricants.

Another interesting phenomenon in which vibration effects may be involved relates to the so-called Timken Abrasion Test. This test is not a film-strength test, but is made to determine the abrasive tendencies of a lubricant at a moderate degree of loading for an extended operating period. Running the machine normally, with the friction lever free or "floating", the wear with E.P. Lubricant *B* was excessive (see Fig. 1). Yet, when the test was made in the same way with the friction lever heavily loaded or locked to prevent movement, the same lubricant showed very low wear. A second lubricant, E.P. Lubricant *C*, showed no change in abrasion

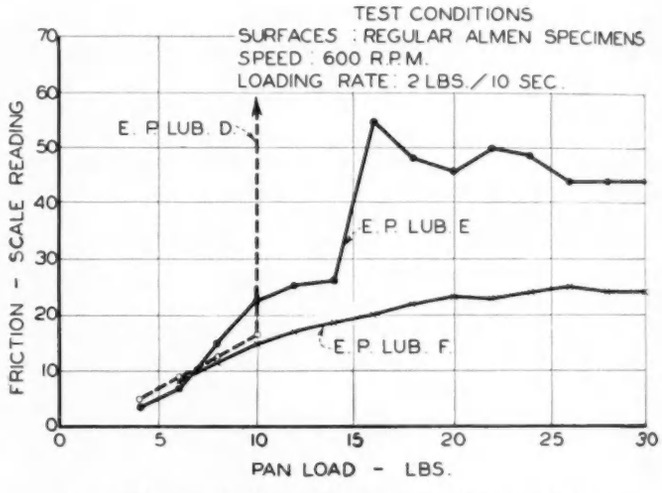

TEST CONDITIONS
SURFACES : REGULAR ALMEN SPECIMENS
SPEED : 600 R.P.M.
LOADING RATE : 2 LBS./10 SEC.
E. P. LUB. D.
E. P. LUB. E
E. P. LUB. F.
FRICTION - SCALE READING
PAN LOAD - LBS.

Fig. 4—Almen Machine—Difference in Frictional Characteristics of Extreme-Pressure Lubricants

whether the friction lever was locked or free. It is believed that differences in abrasion just discussed depend primarily on the film strengths, even though the test is conducted at a load far below the nominal film strength of the product.

Fig. 2 shows the effect of concentration of compounding on bearing wear as obtained in the Timken machine using steel on brass specimens. It will be noted that concentrations of Compound *A* up to about 3 per cent were detrimental in this test while 5 per cent markedly reduced the wear. Very similar results were obtained with Compound *B* in smaller concentrations, while Compound *C* showed no such negative effects. These deleterious effects of small concentrations persist even in the presence of further addition agents of high film-strength values.

While it is realized that the steel-on-brass wear data are rather extraneous as related to the film strengths of gear oils, they do serve to illustrate a type of problem to be considered in the testing of lubricants at high unit loads such as prevail, for example, in worm gearing. Such indications also serve to warn of the possible deleterious effects of dilution or mixing products.

Fig. 3 shows a negative oiliness result somewhat similar to

those shown on Fig. 2, except that it is a load phenomenon rather than one of concentration of the compounding material.

Typical Almen machine results obtained with the manufacturer's recommended operating procedure are shown on Fig. 4. It is shown that, while E.P. Lubricant *E* successfully passed the test, that the friction becomes very high at relatively light loads. With E.P. Lubricant *F*, the friction remained low for the entire test, yet both lubricants carried the full load of the machine. It is noted that E.P. Lubricant *D* showed low friction values up to a load of 10 lb., at which load the bearings seized. These data duplicate those previously reported by other investigators, but are included to emphasize that friction-reducing value should not be ignored in the testing of extreme-pressure lubricants.

Fig. 5 is of special interest as it shows how the results for different types of lubricants can be altered in the same machine by using different operating conditions. The upper chart shows the results obtained with mineral oil and with E.P. Lubricants *G* and *H*, using the regular Almen machine testing procedure. When this procedure is altered slightly, either by using a slower rate of loading shown in the chart to the right of the first one or by increasing the rubbing speed as shown directly below, the results are reversed. A reversal of results also was obtained with different types of steel test surfaces. The third upper barograph from the left shows results obtained with regular Almen test pins and Timken test blocks. It is noted that E.P. Lubricants *G* and *H* were both about equal in this test, while straight mineral oil showed poor results. The results shown in the fourth upper chart from the left were obtained with the Almen operating procedure using Timken test blocks and special-steel test pins of about the same hardness as the Timken blocks. Lubricant *G* excelled Lubricant *H* in this test and also was superior for the conditions directly below which were the same except for a higher speed. With such contradictory data as these, it is difficult to evaluate the lubricants being considered. These findings clearly illustrate the need for adequate gear-set test data for selecting proper operating conditions for film-strength testing machines.

Aside from the nationally known extreme-pressure-lubricant testers, such as those previously referred to, it may be well to mention that many of the laboratories have constructed devices of their own design for the study of extreme-pressure properties. One of the machines designed and used by our laboratories for this purpose is a small-size worm-gear-lubricant testing machine. With this device practical data can be obtained in regard to film strength, friction-reducing value, gear wear, and stability and life of lubricants for various gear-metal combinations. It was selected for extreme-pressure testing since the sliding friction of hypoid gears and of steel-on-steel worm gears is the basis of the extreme-pressure-lubricant problem. This machine has been correlated with full-scale gear-set results for steel-on-bronze worm-gear operation, and is therefore adaptable to the study of the principal gear-lubricant problems existent today. The machine consists of a worm reduction gear loaded by a prony brake that is held automatically at constant load; it is driven by a motor supported in a swinging frame so that the power input can be measured and the efficiency calculated. Several thousand tests have been made with this machine operated under conditions of high load to obtain data on gear scuffing and other performance characteristics. It is interesting to note that many E.P. lubricants now on the market will not pass this test.

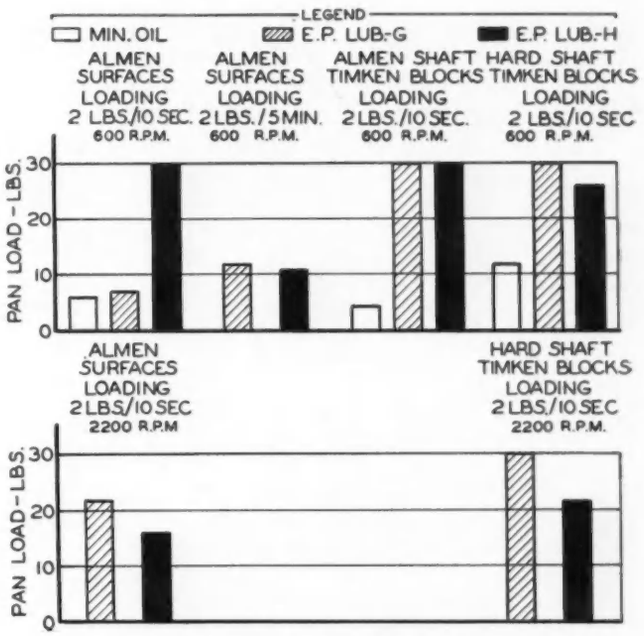

LEGEND
MIN. OIL
E.P. LUB.-G
E.P. LUB.-H
ALMEN SURFACES LOADING 2 LBS./10 SEC. 600 R.P.M.
ALMEN SURFACES LOADING 2 LBS./5 MIN. 600 R.P.M.
ALMEN SHAFT TIMKEN BLOCKS LOADING 2 LBS./10 SEC. 600 R.P.M.
HARD SHAFT TIMKEN BLOCKS LOADING 2 LBS./10 SEC 600 R.P.M.
PAN LOAD - LBS.
0
10
20
30
ALMEN SURFACES LOADING 2 LBS./10 SEC 2200 R.P.M
HARD SHAFT TIMKEN BLOCKS LOADING 2 LBS./10 SEC 2200 R.P.M.
PAN LOAD - LBS.

Fig. 5—Almen Machine—Effect of Metals and Operating Conditions on Load-Carrying Capacities

Fig. 6 shows the results obtained with two different extreme-pressure lubricants in several laboratory tests. It is shown that Lubricant *I* has a higher film strength than Lubricant *J* in the Timken machine, but is inferior in the Almen machine. Lubricant *I* was considerably superior to Lubricant *J* in the worm-gear machine at 275 deg. fahr. It will be noted that Lubricant *J* allowed much the greater friction for steel-on-bronze gears and also allowed scuffing with steel-on-steel gears. At 175 deg. fahr. both lubricants showed the same friction loss in the worm-gear machine for steel-on-steel surfaces but, in the wear data shown on the right side of the chart which were taken from the same steel-on-steel runs in which the friction results for these two oils were equal, it is shown that the gears and bearings were wearing nearly twice as fast with Lubricant *J* as with Lubricant *I*.

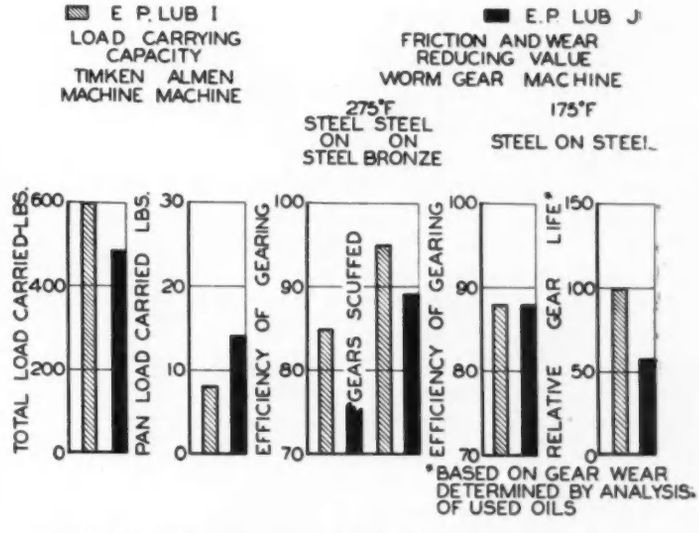


Fig. 6—Performance Data for Extreme-Pressure Lubricants—Comparing Timken, Almen, and Worm-Gear Machines

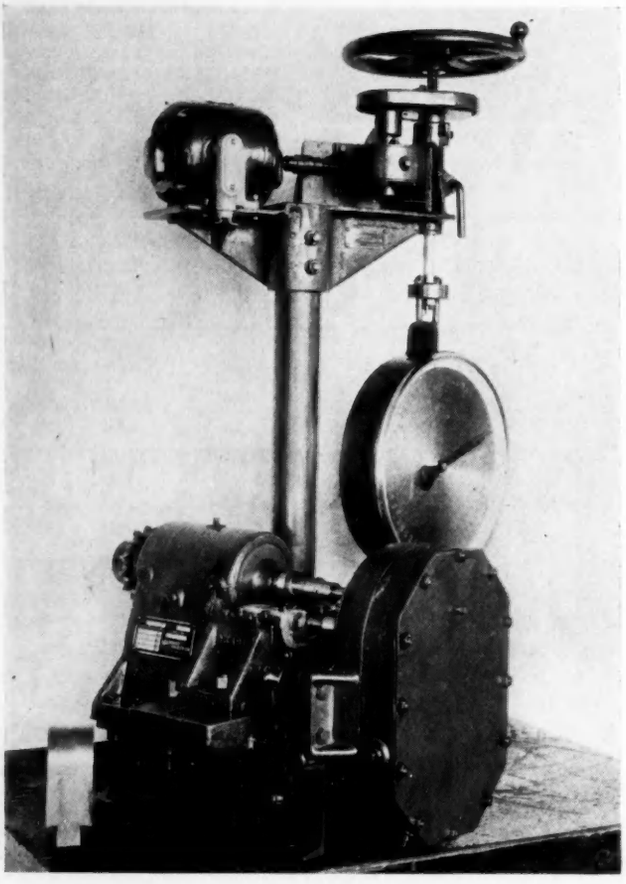


Fig. 7—S.A.E. Extreme-Pressure-Lubricants Tester

In view of the difficulties of correlation with the machines examined by the Bureau of Standards, the Bureau, acting under the sponsorship of the S.A.E. subcommittee, designed and built the first model of the S.A.E. Extreme-Pressure-Lubricants Tester. Subsequent thereto, the machine has been redesigned and improved by W. S. James of the Studebaker Corp. and by T. C. Smith of American Telephone and Telegraph Co. Due credit should be given to these gentlemen for producing the present design, which is now distributed in some 18 laboratories throughout the United States for cooperative test work.

The most important feature of the S.A.E. Tester, shown in Fig. 7, is the fact that both of the rubbing surfaces are continuously in motion during testing. These surfaces are the outsides of Timken test cups, the same as are used for the Timken machine, which are mounted on two shafts that are caused to rotate at different speeds (see Fig. 8). The cups are pressed together by means of a screw and lever system, and the load is shown on the large dial scale. The test lubricant is contained in an oil box that holds the lower shaft, with this shaft and the lower test specimen continuously dipping in the lubricant being tested. All parts of the leverage system operate through knife edges except the oil-box trunnion or frame which is hinged through a piece of flexible shim steel. The criterion of the machine is the load at which the test cups start to scuff, which load is shown by lines appearing on the surfaces.

The S.A.E. Tester is equipped with means for varying the operating conditions over a wide range. It is possible to change the speed of the machine by changing the speed of the driving motor. The loading rate may be regulated by the use of various sizes of change gears to drive the loading unit. The ratio of speeds of the test cups is dependent on which of several sets of gears are used to transfer power from the upper shaft to the lower shaft.

It is believed that the new machine constitutes an approach to the gear problem that is more sound fundamentally than any of the other machines that have been used. In the older machines one or more of the test surfaces was always stationary while, in the S.A.E. Tester, both surfaces are in motion and a combination of sliding and rolling friction approaching that of gearing may be achieved.

In spite of this progress, however, the problem is by no means solved, and a true solution can be achieved only by correlating the S.A.E. Tester with actual service tests. It seems very possible that, even though a correlation may be achieved, it will require a large number of operating conditions to include all of the ramifications of duty of a commercial extreme-pressure gear oil. The manufacturer of gear lubricants for general sale must consider more than merely the load-carrying capacity of his product. He must also consider lubricating value for the several metal combinations found in the different types of gear sets; stability in storage and in service; corrosivity; wear-reducing value of the lubricant; general serviceability such as foaming, odor, gear-shifting characteristics; and so on.

It is, therefore, easily seen that, in view of the contradictory results obtained in other machines, such contradictions might be obtained with the S.A.E. Tester. It is believed, however, that the problem of correlating the laboratory data with service data will be simplified by the use of this new machine.

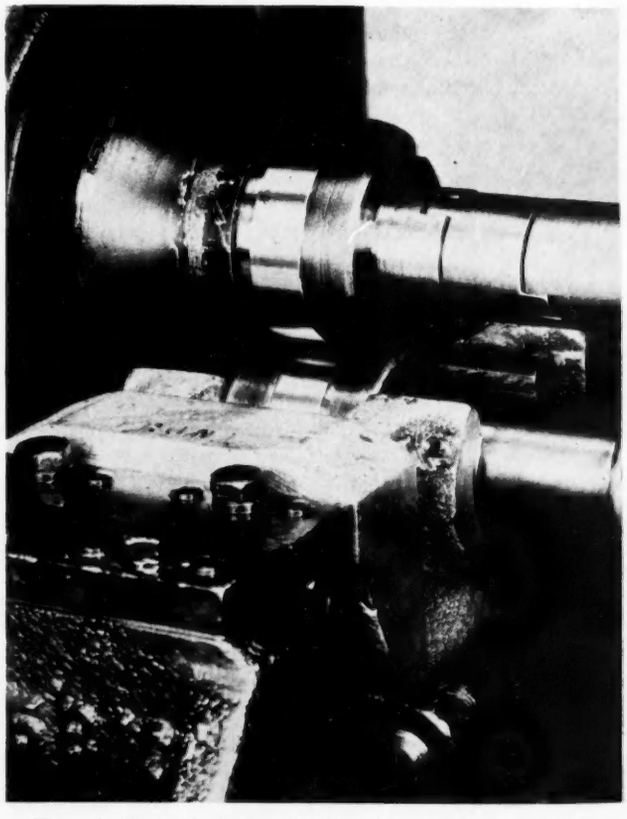


Fig. 8—Test Surfaces—S.A.E. Extreme-Pressure-Lubricants Tester